Aalto University

Bjørnar Vattøy

Techno-economic evaluation of heat-driven cooling solutions for utilization of district heat in Aalesund, Norway.

Aalto University
School of Engineering

Aalto University, P.O. BOX 11000, 00076 AALTO
www.aalto.fi

Abstract of master's thesis

Author Bjørnar Vattøy	
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Thesis advisor(s) Irene Vik	
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Abstract

This study is aimed to evaluate the techno-economic feasibility of implementing heat driven cooling technologies in buildings connected to Tafjord Kraftvarme's district heating network in Aalesund, Norway. Heating and cooling demands were found by projecting two 4000 m^2 office buildings according to Passive House and Low Energy Building criteria, within the frame of the energy requirements in the TEK17 building regulations (Standard Norge, 2012) (Norwegian Building Authority, 2017). Suitable cooling and heating equipment, both electrical and heat driven, were dimensioned based on the peak cooling load of the projected buildings, and technical and economic information obtained from the distributors of the equipment.

LCOE analysis shows that the heat driven cooling solutions could be able to compete economically, in variable extent, with the electrically driven solutions given relatively low heating demand or by applying investment subsidies or price reduction on district heat for cooling purpose. The desiccant cooling solution could even compete with the electrical driven solution even without subsidies or price reduction on DH for cooling. This is mainly because of its enhanced heat recovery reducing the heating demand. The absorption cooler on the other hand, has both a higher consumption and higher power input of district heat while running, and is therefore less competitive without subsidy or price reduction on DH for cooling. In the building cases explored, the absorption cooling solution requires either subsidy or price reduction on DH for cooling to compete with the electric chiller and district heat solution, while it require both to come close to compete economically with the heat pump solutions. With increasing heating demand the heat driven solutions, which use district heat as their heat source, become less competitive compared to the heat pump solutions. This is because, with the mild winters in Aalesund, the heat pumps can run with a COP of 2-3 while the COP of district heat it is considered to be 1.

Other important factors that is not covered by the LCOE analysis is the reliability and environmental aspect. None of the heat driven cooling solutions use any environmentally unfriendly refrigerants, and the maintenance of the cooling machines are minor. The lifetimes of the machines are estimated to be 20 years for the desiccant and 40 years for the absorption cooler, compared to 15 years for the heat pumps and electric chiller. The heat driven cooling solutions can therefore be considered more reliable both in terms of regulations on refrigerants and on maintenance and lifetime. An additional important factor is that compared to the heat pump solutions, the heat driven cooling solutions with district heat to cover the heating demand can replace large quantities of electricity consumption with low-grade thermal energy.

Considering both the economic, environmental and reliability factors, the heat driven cooling solutions could be a viable option and should be considered when implementing heating and cooling equipment in buildings connected to the district heating network in Aalesund or other locations with similar climate.

Keywords District heating, heat-driven cooling, Aalesund, techno-economic analysis.

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Abbreviations

LCOE	**Levelized Cost of Energy**
DH	**District Heating**
ISO	**International Organization for Standardization**
GWh	**Gigawatt hours**
DEC	**Desiccant cooling**
COP	**Coefficient of performance**
CFCs	**Chlorofluorocarbon**
ARI	**Air-Conditioning and Refrigeration Institute**
DW	**Desiccant wheel**
HE	**Heat exchanger**
DUT	**Dimensioning outdoor temperature**
PH	**Passive house**
LEB	**Low energy building**
AHU	**Air handling unit**
SFP	**Specific fan power**
HP	**Heat pump**

1 Introduction

One of the greatest challenges faced by the present and future generations is understanding how to manage the large amount of wastes produced by the society. Minimization of waste production and recycling of larger portions of the waste material are some of the approaches being used presently. However, considerable amounts of undesirable waste products for recycling are still being produced and this has instilled interests for other solutions than simply landfilling (Council, 2016).

In urban areas with high energy demand, waste to energy through incineration is often considered a desirable option. This solution has been seen as a viable option, producing energy in the form of electricity and heat, and thereby solving environmental related issues caused by the landfilling due to the production of unrecyclable wastes. Incineration reduces the volume of the landfilled wastes thereby reducing the demand for landfilling space. It also helps to eliminate methane gas production from waste treatment processes and neutralizes hazardous wastes by releasing the hazardous contents as flue gas which is then collected with several cleaning technologies (Bank, 1999).

1.1 Tafjord Kraftvarme's Waste to Energy plant

Tafjord Kraftvarme's Waste to Energy plant in Aalesund, Norway is a co-generation plant that utilizes the heat from burning wastes to produce electricity and district heat. The plant is environmentally certified according to ISO 14001:2015 and in energy management according to ISO 50001. The wastes used in this plant are a mix of municipal waste, industrial waste and hazardous waste from the local medical institutions. Electricity produced from this plant is transmitted to the electricity grid and sold through Aalesund municipality, while the district heat produced is distributed through a district heating network and sold to local customers (schools, shopping center and residential buildings) (Irene Vik, 2017).

The plant plays an important role in the local community, serving as an economical and environmental friendly solution which satisfies the local energy demands. With Aalesund municipality as the major shareholder in the Tafjord Group and the municipalities Nordal and Orskog as minor shareholders, it contributes economically to the local society through yearly dividends (Tafjord, 2018). By burning hazardous waste from local medical institutions, the waste is neutralized locally, avoiding costly and unsustainable transportation. The waste is going through a quality control and churning at an external company before it is delivered to the plant. After utilizing the waste for thermal energy, the slag produced is collected by the same external company, which extracts the metals and use what remains as surface for landfills (Irene Vik, 2017).

On some days during the winter the demand of district heat peaks and all production is utilized. However, during most of the year and especially during the summer period, production exceeds demand, leading to large quantities of excess heat being cooled away in the ocean. In 2015, the produced thermal energy that could have been utilized for district heating was 236.5 GWh. Additionally, about 22.5 GWh could have been recovered from the wet scrubbers. Since the delivered district heat was 110GWh* (Network losses included), hence there is a significant need to increase the utilization of the wasted district heat produced, especially during warmer months (Irene Vik, 2017).

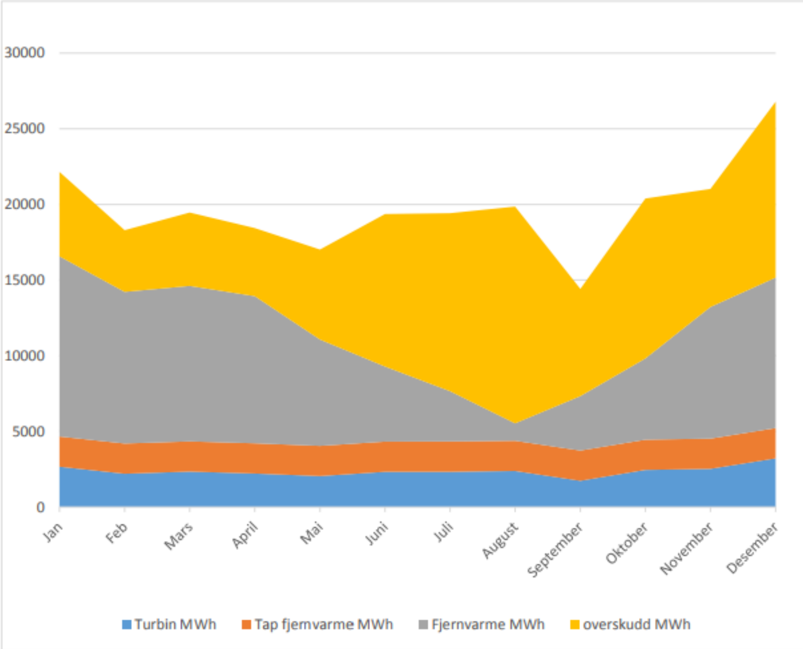

Figure 1.1: Energy production at Tafjord Kraftvarme's WTE plant for 2015 (Irene Vik, 2017).

Till this moment, the district heating network has mainly been used to provide conventional district heating services to customers who require them and are located close to the district heating network. With the demand for these services driven by low outdoor temperatures, the seasonal demand of district heat from the plant has become unbalanced, with peak periods during the winter where all the available energy is used and excess heat during warmer periods. To better balance the demand and reduce the excess heat wasted, a change in approach is necessary. This thesis includes developing improved decision-making criteria to benchmark possible district heating projects as well as researching and implementing technologies that would use district heat during "off-seasons" for regular heating demand (Irene Vik, 2017).

1.2 Scope of work

The scope of work is to review different options for heat-driven cooling that could be applied in Aalesund to utilize excess district heat during warmer months with lower heat demand. The heat-driven technologies are to undergo a technical-economic evaluation where they are benchmarked against electrically driven technologies.

1.3 Limitations

- During the planning and design of the buildings for this thesis, the focus has been on fulfilling the energy and daylight requirements given in TEK 17 and NS 3701:2012. Other requirements, such as fire safety or installation of person lifts have not been considered.
- Hourly temperatures from the SIMIEN climate data base are used in the energy simulations in SIMIEN. Temperatures can fluctuate based on geographical location, even within the same city.
- No reliable data is available in hourly resolution for humidity in Aalesund. Humidity data from the local airport located in the neighbor municipality is therefore used to determine the COP of the desiccant cooling machine.

2 Heat-driven Cooling technologies

Heat-driven cooling systems produce cooling by utilizing low-grade thermal energy such as district heat, rather than electricity (Narayanan, 2017). Given that the current configuration of the customer base for district heat in Aalesund results in large amounts of excess heat during warmer periods, a switch from electrically-driven to heat-driven cooling solutions could have several positive effects. With the utilization of otherwise wasted energy, large amounts of energy could be saved. Undesirable electricity demand peaks enhanced by electrically driven cooling systems would decrease in magnitude. Also, the environmental unfriendly refrigerants used in heat pumps and electric chiller could be reduced or eliminated, depending on the technology used (Pradeep Bansal, 2012).

Based on the literature review, absorption and desiccant cooling were identified as the most promising heat-driven cooling technologies available for implementation into Tafjord Kraftvarme's district heating network. These cooling technologies can be divided into closed and open cycle systems, with absorption chillers in the open cycle category and the desiccant cooling systems in the closed cycle category. The closed cycle system chillers produce chilled water that can be used for cooling purposes, while for the DEC systems, the process air is treated directly through a combination of a dehumidifier, a sensible heat exchanger, and evaporative coolers (Narayanan, 2017) .

When determining the most suitable cooling solution, the cooling load requirement for the building is a dominant factor, along with coefficient of performance and price. The cooling load is the amount of heat that needs to be removed by the air conditioning system to obtain the desired conditions. This can be divided into sensible and latent load. The sensible load is the load used to reduce the dry bulb temperature of the air, while the latent load is load required to reduce the moisture content of the air (Narayanan, 2017) (K. Daou, 2006) (J. Steven Brown, 2014).

2.1 Absorption Cooling

The absorption cooling systems have a similar buildup as the vapor compression systems, with the main difference being that the compressor is replaced with an absorber and a generator, which allows low grade heat energy to be used as the principle driver instead of mechanical energy. This reduces the electricity consumption drastically. However, the COP of the absorption cooling systems are usually lower than the one in vapor compression systems, meaning that their economical sustainability is strongly dependent on the electricity price and the access of low cost thermal energy (Corrada, 2015) (J. Steven Brown, 2014).

There are two working fluids in the absorption chillers, the absorbent fluid and the refrigerant. The function of the absorbent fluid is to absorb the evaporated refrigerant on the low-pressure side, forming a solution that can be pressurized before it is desorbed in the generator. The conventional working fluid pairs are ammonia/water with ammonia as the refrigerant and water as the absorbent, and water/lithium bromide where water serves as the refrigerant and lithium bromide as the absorbent. For air conditioning the water/lithium bromide chillers are the most commonly used. However, using water as the refrigerant limits the minimum temperature to 0 °C, meaning that these chillers are not very suited to serve as industrial fridge and freezing systems. For these systems, the ammonia/water chillers with the possibility of going down to -30 °C would be the preferred option (J. Steven Brown, 2014) (Corrada, 2015).

The absorption chillers come in several configurations and are categorized as half, single, double and triple-effect systems. The double and triple effect chillers have a significantly higher COP than the single-effect chillers. However, while the classic single-effect chillers usually require driving temperatures of 80-100 °C, the double and triple-effect chillers typically require driving temperatures of above 130 °C and 180 °C, and can therefore not be driven by district heat, of which the temperature usually lays around 100 °C or lower. The half-effect chillers on the other hand, are designed for cases where the heat sources are not able to reach a high enough temperature to run a single-effect chiller. The principle of the half-effect chillers is to have two heat lifts by adding an additional absorber and generator. This way the chillers can be run by a lower driving temperature, but it also results in a reduced COP. While the single-effect chillers are reported to have a maximum COP of 0.85, the half effect are restricted to about 0.45 due to the double heat input (Gomri, 2010) (Narayanan, 2017) (Magnus Rydstrand, 2004).

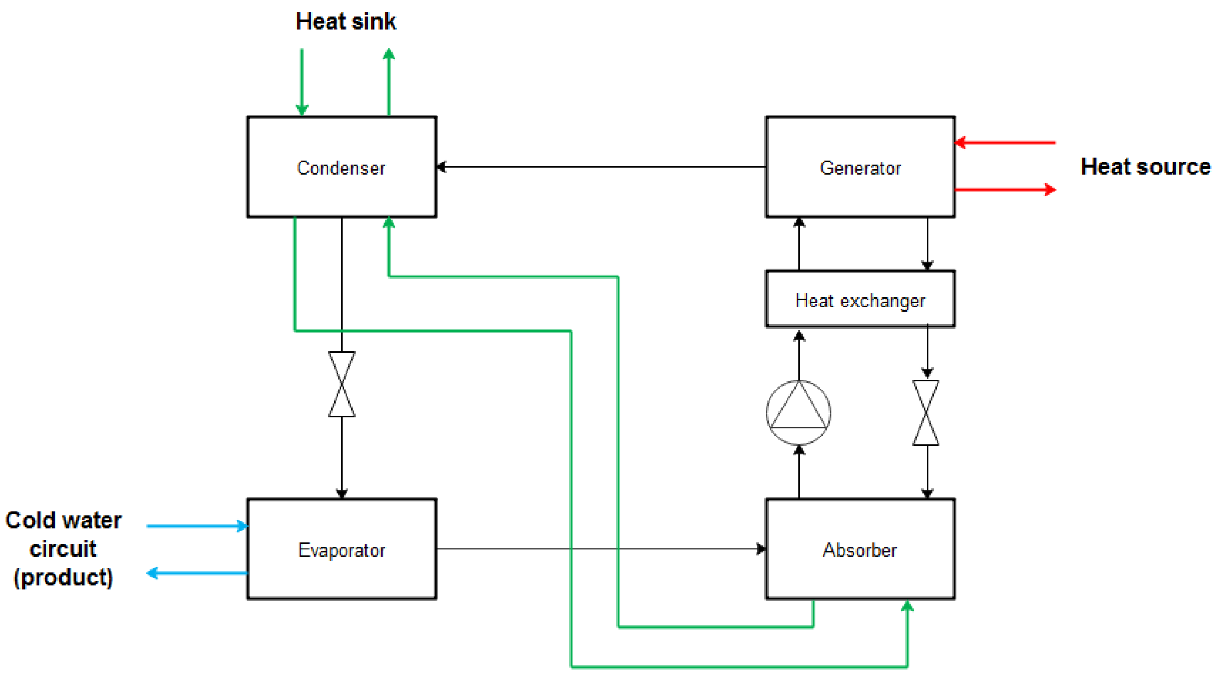

Figure 2.1: Single effect ammonia/water absorption chiller.

2.2 Desiccant Cooling

The concept of desiccant cooling is to use a desiccant material to remove moisture from the incoming airstream through adsorption, reducing the latent load, before adding water through evaporation to cool the air adiabatically (Magnus Rydstrand, 2004). For the DEC system to work continuously, the desiccant material must be regenerated before starting a new cycle. This is done by heating the desiccant material to its regeneration temperature, where the adsorbed water vapor is driven out of the material and the desiccant is re-activated and ready to adsorb more vapor in the next cycle (K. Daou, 2006).

The DEC systems can be divided into liquid and solid desiccant systems. For the liquid systems, the incoming airstream is put in direct contact with a liquid desiccant that absorbs some of the airstream's water content, while for the solid systems, the air is put in direct contact with a solid desiccant.

There are several kinds of solid desiccant systems, but in this paper the focus will be on the rotary desiccant wheel systems. In these systems, the incoming air is exposed to one side of a desiccant wheel, which adsorbs the moisture of the airstream, while at the same time the heated exiting air regenerates the other side of the wheel (J. Steven Brown, 2014). The reasoning behind this decision is that the rotary desiccant wheel systems, even though its prevalence is not very high, are the most popular and commercialized of the desiccant systems (D. La, 2010) (Magnus Rydstrand, 2004).

Rotary desiccant wheel systems

Combining the technologies of desiccant dehumidification and evaporative cooling, using low grade thermal energy as the driving force and water as the refrigerant, the rotary desiccant wheel systems have several advantages over the vapor compression systems. The usage of CFC refrigerants is avoided, the electricity consumption is lower and it can control both the temperature and the humidity. Additionally, the construction and maintenance are simple (Ghassem Heidarinejad, 2010). When compared to the other desiccant cooling systems, the rotary wheel is preferred because of its high capacity, low pressure drop over the wheel, low dew point, its ability to work continuously and it is less subject to corrosion (D. La, 2010) (Narayanan, 2017).

Given the Nordic climate in Aalesund, the warm and humid days with high cooling requirements are limited to shorter periods, mainly during the summer months and might not justify the investment of a rotary desiccant system by itself. However, during colder periods the desiccant wheel can be used to enhance the heat recovery through adsorption of moisture in the exhaust air. This way the demand for district heat during the winter period is reduced, while it is increased during the summer (Magnus Rydstrand, 2004).

Operation processes

Figure 2.2 illustrates the rotary desiccant wheel cycle as it was patented by Pennington in 1955. It also goes under the name ventilation cycle since there is no mixing or recycling of air. Solely ambient air is used in the process airstream (1-4) and solely return air is used in the regeneration airstream (5-9) (D. La, 2010) (Ghassem Heidarinejad, 2010). According to (Brum, 2014), the ventilation cycle's configuration has been extensively tested and analyzed both experimentally and numerically. In the examples given in the report, the experiments with various desiccant materials and outdoor conditions, resulted in thermal COPs ranging from 0.28 to 0.78, while a numerical simulation under ARI conditions and with ideal components gave a COP_{th} of 1.04 (Brum, 2014).

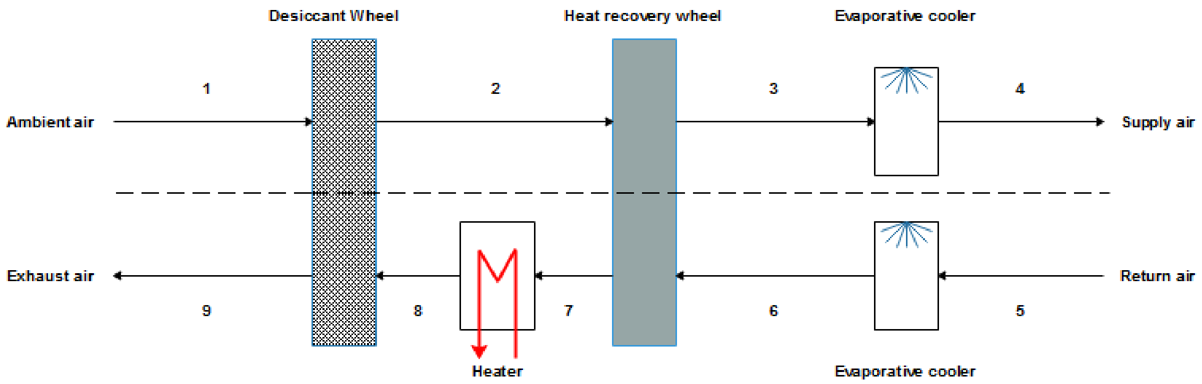

Figure 2.2: Pennington cycle.

Starting from point 1 in the figure, ambient air from outside the building is blown towards the desiccant wheel. When the air passes through the slowly rotating DW in process 1-2, moisture is adsorbed by the desiccant material, which reduces the latent load and leads to an increase in the temperature due to the adsorption heat effect. The air in 1-2 is also exposed to additional heat through heat exchange from the regeneration side (8-9). In the next process 2-3, the air is precooled by a rotating heat recovery wheel that exchanges sensible heat from the process to the return air. In the last stage before entering the building 3-4, the process air passes through a direct evaporative cooler that brings the air to the desired supply air temperature and humidity. Simultaneously, the return air from the building passes through the desiccant system in the opposite direction from point 5 to 9. The return air is first treated by a direct evaporative cooler, where the return air is humidified and cooled close to the wet bulb temperature before entering the heat recovery, which improves the cooling performance. In the next steps, 6-7 and 7-8 the air is heated to the desiccant material's regeneration temperature. First a pre-heating process through the heat recovery wheel and then it is brought up to the regeneration temperature by the external heat source. In its final stage

8-9, the return air regenerates the desiccant material before being exhausted in point 9 (Narayanan, 2017) (D. La, 2010) (Ghassem Heidarinejad, 2010).

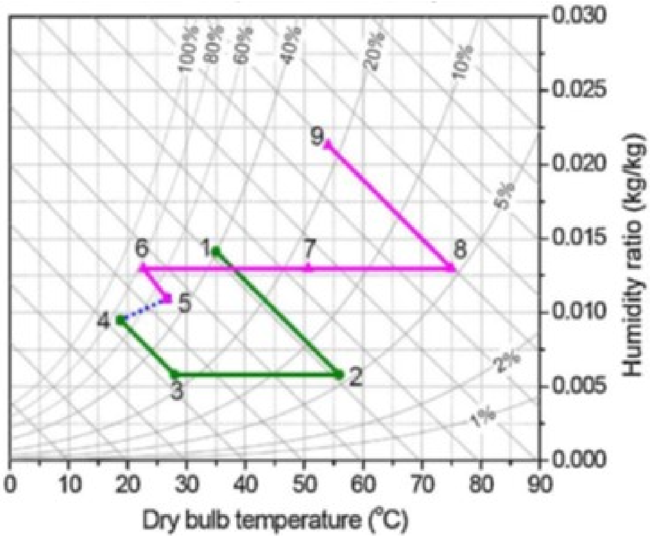

Figure 2.3: Pennington cycle, psychrometric chart (D. La, 2010).

Alternative cycles

In some cases, it could be inconvenient or undesirable to use ambient air for the process air and return air for the regeneration airstream because of the outdoor conditions. In these cases, alternative cycles could be used. The simplest solution is to modify the ventilation cycle by changing the air flows, while the more advanced involves adding extra equipment such as heat exchangers and/or desiccant wheels, or changing from direct evaporative coolers to cooling coils and cooling towers.

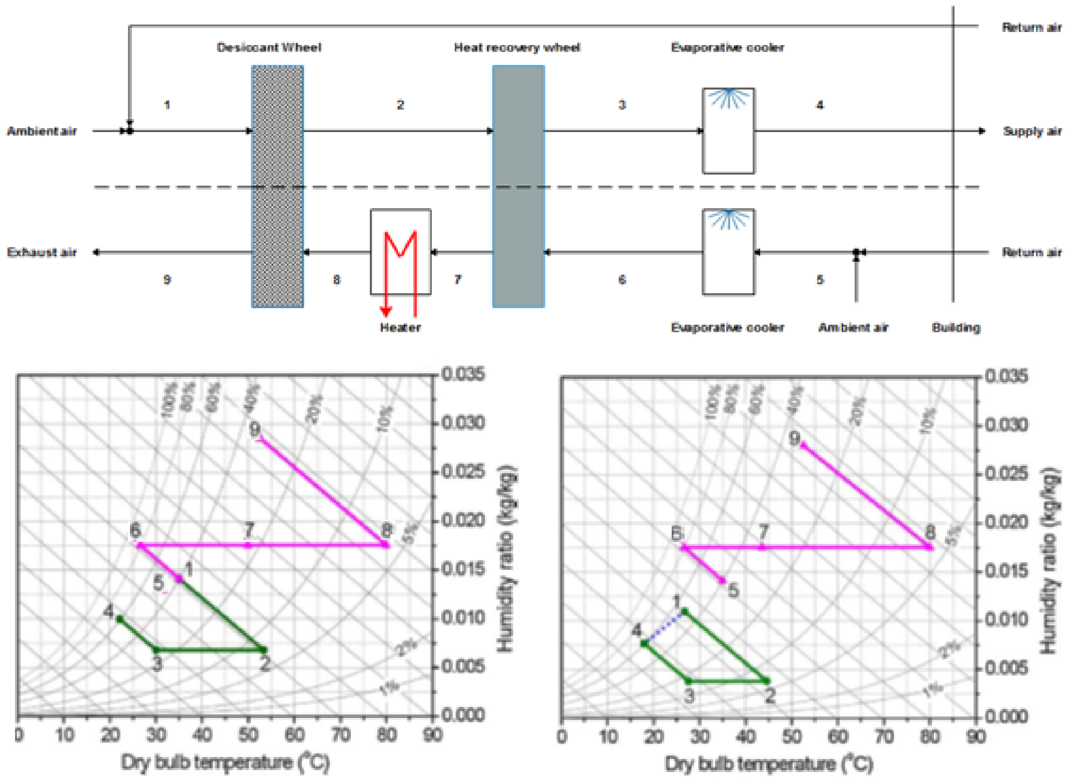

Figure 2.4: A rotary desiccant setup with ventilation, recirculation, makeup and mixed mode. Psychrometric charts of the makeup cycle (left) and recycling cycle (right) (D. La, 2010).

The schematic in figure 2.4 shows a setup where the desiccant wheel system can be set to ventilation, recycling, mixing or makeup mode. However, the systems can be built as solely ventilation, recirculation or makeup cycle as well. The makeup cycle uses ambient air for both the process and regeneration airstream. Since the return air usually is less humid and at a lower temperature than the ambient air, the cooling load and coefficient of performance are generally lower compared to a ventilation cycle. For the recirculation cycle only return air is used as process air, while the regeneration airstream is supplied with only ambient air. According to (D. La, 2010) the $\mathrm{COP_{th}}$ of the recirculation cycle is usually limited to no higher than 0.8 due to the return air's relatively low temperature and humidity ratio. Additionally, the lack of fresh air is a major disadvantage. In the mixed mode purposed in figure 2.4, either the process, regeneration or both airstreams consist of a mix of ambient and return air (D. La, 2010) (Ghassem Heidarinejad, 2010). In 2010, Bourdoukan reported that under conditions with low outdoor humidity a ventilation cycle can have higher cooling capacity and $\mathrm{COP_{th}}$ than the recirculation or mixed cycle (P. Bourdoukan, 2010).

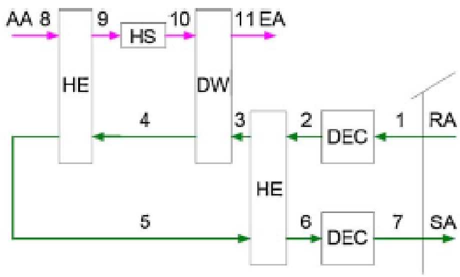

Figure 2.5: Dunkle cycle (D. La, 2010).

Other more advanced one-step rotary desiccant cycles are the Dunkle, SENS, REVERS and SENC cycle. The Dunkle cycle utilize both the recirculation cycle's ability to provide a relatively large amount of cooling capacity, and the ventilation cycle's ability to provide cold air with relatively low temperature for the heat exchanger. This is done by adding an extra heat exchanger as shown in figure 2.5. Since this cycle uses solely return air as for the process airstream, it has the same disadvantage as the recirculation cycle, with lack of fresh air.

The figures underneath shows the SENS and REVERS cycles. In these cycles, the direct evaporative coolers are replaced with a cooling coil and a cooling tower. Additionally, the SENS cycle also incorporates an additional heat exchanger. In 2014, Brum reported that simulations under ARI conditions with ideal components gave a $\mathrm{COP_{th}}$ of 2.58 for the SENS cycle. However, according to (D. La, 2010), the SENS cycle is blocked by its complexity. The REVERS cycle is a simplified version of the SENS cycle where the additional heat exchanger is removed and the air from the cooling tower is forwarded straight to the remaining heat exchanger. The removal of one heat exchanger would increase the temperature of the process air entering the cooling coil, and thereby reduce the cooling effect. On the other hand, avoiding the use of ambient air in the regeneration process would reduce the amount of external heat required. Thus, the REVERS cycle requires less external heat than the SENS cycle, but it also has a lower cooling effect. An additional configuration is called the SENC cycle. It has one heat exchanger similar to the REVERS cycle, but it uses ambient air as the input to the remaining heat exchanger similar to the SENS cycle. The SENC cycle has been experimentally tested and analyzed with a reported thermal COP ranging from 0.61 to 1.97 with regeneration temperatures from 54.1 to 90.4 Celsius (D. La, 2010) (Brum, 2014).

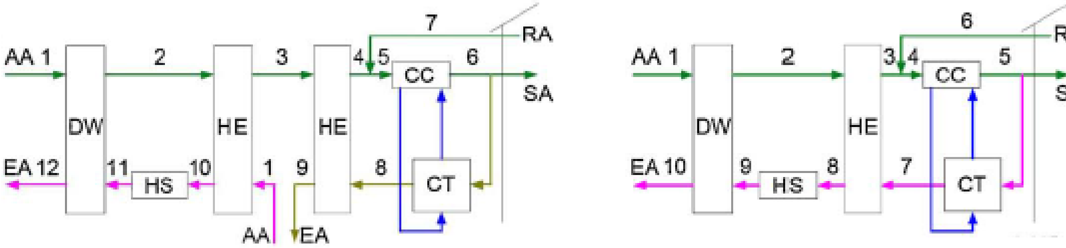

Figure 2.6: SENS cycle on the left-hand side and REVERS cycle on the right (D. La, 2010).

Staged regeneration

Another way to improve the rotary desiccant system is through staged regeneration. This can be done either by staging the regeneration of one desiccant wheel or by adding additional DWs and HEs. For the one-wheel staged regeneration, the regeneration side of the wheel is split in two sections, one for pre-regeneration and one for complete regeneration. After the regeneration airstream leaves the HE, a fraction of the stream is sent directly to the pre-heating fraction of the desiccant wheel, while the other fraction is directed towards the external heat source for further heating before it enters the desiccant wheel and complete the regeneration. This method increases the COP by sacrificing cooling capacity and, according to (D. La, 2010), it reduces the effectiveness and size requirements of the HE, and thereby the cost and size of the system.

The multiple-stage rotary desiccant systems (additional DWs and HEs) are introduced to counter the chain effect created by the adsorption heat release during the dehumidification process. The chain effect starts with the mentioned heat release, increasing the sensible heat and reducing the relative humidity of the process air. This results in a reduction in the vapor difference, which limits the dehumidification ability, since the vapor difference is the main driving force for dehumidification. Thus, especially for high humid climates, a much higher regeneration temperature is required to gain the dehumidification capacity desired. (D. La, 2010) suggest that theoretically, by staging the dehumidification with infinite DWs and HEs, the dehumidification process should become close to isothermal. Then the regeneration temperature for the system would be the minimum and the heat recovery measures would consume less heat for regeneration. The results would be a significant increase in both the applicability and COP_{th} (D. La, 2010). Recent studies have been focusing on developing and testing these multi-staged systems. This has been documented by (T.S. Ge, 2015), which states that there has been both theoretical and experimental results showing that, compared to one-staged systems, two-staged rotary desiccant systems have the merits of higher COP_{th} and lower regeneration temperature.

Desiccant materials

The frame of the desiccant wheel consists of small parallel channels made from supporting material, which is coated with a thin layer of desiccant material. The channels come in various structures such as sinusoidal, triangular or honeycomb (hexagonal) (Ghassem Heidarinejad, 2010) (Chung, 2017). The characteristics of the desiccant material used to impregnate the wheel plays a significant role on the performance of the system, and the development of desiccant materials have therefore been subject to extensive research. The most conventional materials used today are silica gel, zeolite molecular sieve, activated alumina, activated carbon and lithium chloride (Narayanan, 2017) (D. La, 2010). According to (D. La, 2010) there are two main principles to follow during the selection of desiccant material. The first being that the material should be easily reactivated and possess a large saturated adsorption amount. Secondly the material should approach the type 1M isotherm, which has been identified as the ideal isotherm shape for solid desiccant materials in cooling applications.

(Narayanan, 2017) highlights silica gel as the most suitable desiccant material for DEC systems. This is based on various parameters such as the micro porous structure resulting in high capacity, no chemical reaction and that the material follows an almost isotherm shape in the adsorption and desorption process. However, (D. La, 2010) reports of composite adsorbents being tested, one with an equilibrium adsorbate uptake that was 2.1 times larger than Silica gel at intake conditions of 25 Celsius and 40% relative humidity. The material also had a regeneration temperature between 60 and 80 Celsius. There are also reports on silica based composites impregnated with LiCl and CaCl2, resulting in 67-145% improvement of the adsorption capacity under classic inlet temperatures of the process air (D. La, 2010). This gives an indication that there is a large potential in the development of desiccant materials that could greatly improve the performance of the rotary desiccant wheel systems.

Performance optimization

Optimizing the DEC systems, both on component and system level is important to improve its performance and competitiveness to other cooling solutions. For the optimization of the desiccant wheel, the factors can be divided into design and operational factors. The operational factors being humidity ratio, temperature and velocity of the process and regeneration air streams, and rotational speed of the wheel. For the design factors, the crucial ones are the thickness and diameter of the wheel, shape and dimensions of the channels, flow configuration, supply to regeneration area ratio, desiccant loading and desiccant layer thickness. It is obvious that the optimization of the desiccant wheel plays a crucial role, but in the system performance perspective the contribution of finding the optimum of other components such as the evaporative cooler and the sensible heat exchanger cannot be overlooked. Modelling of heat and mass transfer in the system is crucial for the optimization, but it can be very difficult due to the lack of knowledge in the porous medium, numerical difficulties in the coupling between heat and mass transfers, and computational time (Chung, 2017) (D. La, 2010).

3 Methodology

With the scope of benchmarking the heat driven cooling solution against electrically driven solutions, a method that evaluates both the technical and economic performance had to be developed. In some cases, the cooling and heating demand are covered by the same machine, such as a reverse heat pump. It was therefore decided to evaluate both the heating and cooling side, as this would give a broader picture of the feasibility of the solutions. A model that would retrieve the cost of each solution covering the cooling and heating demand of a building in Aalesund over a set period was considered. However, since the cooling technologies have different lifetimes, this method was considered inaccurate and it was therefore decided to use a modified levelized cost of energy method instead (Bethel Afework, 2018). In the method used, the costs related to each solution over the cooling equipment's lifetime are summed and divided by the total heating and cooling demand over the same period.

3.1 Design of reference buildings

To perform a techno-economic evaluation on whether heat driven cooling could be a viable cooling solution for buildings in Aalesund or not, the heating and cooling demand for potential buildings had to be determined. Lack of available cooling demand data from existing buildings in Aalesund made it necessary to design and project buildings according to Norwegian building standards, to simulate and obtain realistic demands and peak loads. It was decided to project and perform the evaluation on an office building, since the density and building rate of office buildings are higher than for other types of buildings, such as schools and hospitals.

In a study conducted by SINTEF Byggforsk in 2011, some of the larger contractors and consultant companies within the building sector in Norway were interviewed regarding their use of energy simulation tools. In the interview, all the consultant companies stated that they mainly used SIMIEN for their energy calculations and control against TEK, cooling-/heating loads, energy labeling and the passive house standard (Tor Helge Dokka, 2011). On the background of this, SIMIEN was selected as the simulation software to project the buildings according to the TEK17 building regulations and the passive house/low energy building standard.

3.1.1 SIMIEN

SIMIEN is a simulation software for calculation of energy demand and evaluation of indoor climate (ProgramByggerne). The software allows the user to create zones that can be connected to other zones, the ground or the free through walls, the floor or the roof/ceiling. The performance of each zone's building parts, such as walls, doors and windows are set by the user. Both in terms of U-value, thickness, and other parameters, such as solar shading of windows, and whether they are open or not. The user also inserts heating, cooling and ventilation equipment in the zones, and determine the constraints for these. It is also possible to change the internal loads, such as internal heat gain from technical equipment, lighting and persons.

During simulations, the building is exposed to the climate in the chosen location or normative values (Oslo), dependent of the type of evaluation/simulation performed. The climate database for each location contains both hourly values throughout the year, which is used for most of the simulations, and the dimensioning outdoor temperatures for summer and winter that is used for summer and winter simulations to dimension the cooling and heating equipment. The DUT can be changed for each location by simply changing the value, while the other climate parameters require hourly data throughout the year for the given parameter.

Evaluations in SIMIEN

Creating a building in SIMIEN gives many possibilities such as evaluating the building against regulations, standards and energy marking. For this thesis, SIMIEN was used to evaluate the building against regulations and standards, dimensioning the heating and cooling equipment, and extracting hourly temperature and demand data for heating and cooling through a full year simulation.

For evaluation against building regulations, a full year simulation is performed with normative (Oslo) climate and normative values for internal loads. The software then creates a report containing the performance of the building and how this complies with the building regulations.

For evaluation against the low energy building/passive house standard, a full year simulation is performed with local climate and standard determined values for internal loads. The software then creates a report containing the performance of the building and how this complies with the passive house or low energy building criteria, depending on which one is selected.

The summer and winter simulations are performed in a different way. In the summer/winter simulation, the building is exposed to several summer/winter days with the given DUTs for the selected location. While the summer simulation is 5 days long, the winter simulation is 3. The cooling/heating equipment is then dimensioned based on the peak capacity required to maintain the desired indoor temperature.

The full year simulation without any evaluation against standards or regulations, gives the yearly performance of the building in a report like the other simulations and hourly values in a text file if selected. The hourly data given in the text file is the data used for further calculations in this thesis. The simulation is performed with local climate and should in theory give similar values as the passive house/low energy building evaluation if the building is built after these standards.

Climate data

In SIMIEN, the default dimensioning outdoor temperatures for Aalesund are 21 Celsius and -13 Celsius. However, for decades it has been a norm among the consultants in the area to use 22,4 Celsius as the DUTs (Arild Bjåstad, 2018). It was therefore decided to use this as DUTs to dimension the cooling equipment. With that in mind it would be preferable to evaluate and possibly change the hourly temperature data as well, but reliable data in the right format was not obtained and it was therefore decided to use the default SIMIEN data.

The figure underneath shows the annual outdoor temperatures for Aalesund and Oslo (normative climate) in hourly resolution. This shows that Aalesund have milder winters and colder summers than the normative climate used in the evaluation against building regulations (Oslo).

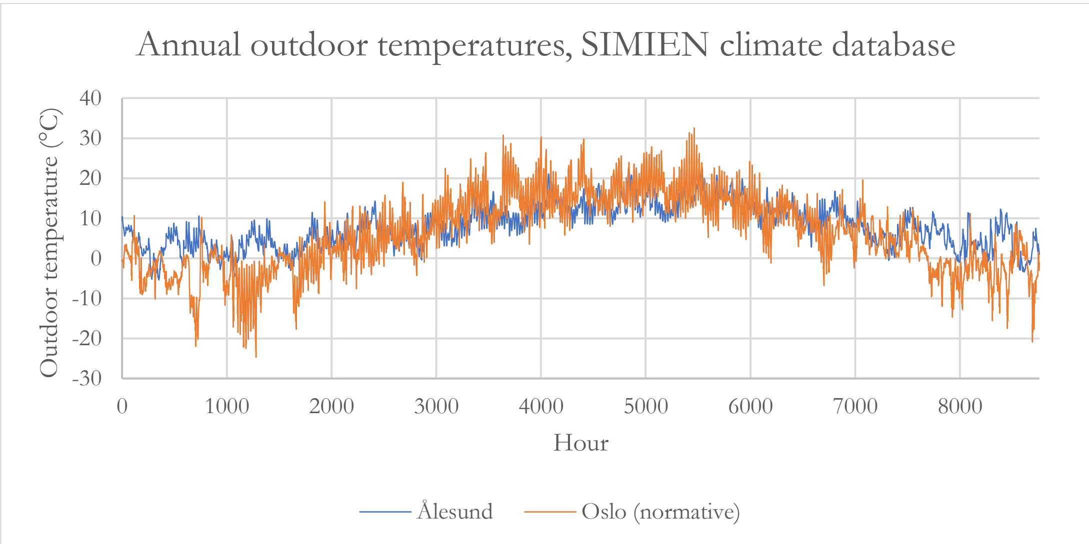

Figure 3.1: Outdoor temperatures for Aalesund and Oslo in the SIMIEN climate database, given in hourly resolution.

3.1.2 Building regulations and standards

In Norway, all new buildings must fulfill the minimum requirements of the regulations on technical requirements for construction works, laid down by the Ministry of Local Government and Modernization (Norwegian Building Authority, 2017). Additionally, there are some stricter standards that is optional, but must be fulfilled for the building to be categorized as a passive house or a low energy building. For non-residential buildings, the criteria for passive houses and low energy buildings are given in Standard Norge's NS 3701:2012 "Criteria for passive houses and low energy buildings – Non-residential buildings" (Standard Norge, 2012).

TEK17 – Regulations on technical requirements for construction works

TEK 17 are the current regulations on technical requirements for construction works in Norway, laid down by the Ministry of Local Government and Modernization in 2017. The purpose of the regulations is to ensure that building projects are "planned, designed and executed on the basis of good visual aesthetics, universal design, and in a manner, that ensures that the project complies with the technical standards for safety, the environment health and energy" (Norwegian Building Authority, 2017).

During the planning and design of the buildings for this thesis, the focus has been on fulfilling TEK 17's energy requirements. The daylight requirements were also considered. However, since the calculation of the daylight requirements became more complicated in TEK 17, the TEK 10 requirements were considered sufficient for this purpose. The TEK 10 guidance states that in rooms for lasting stay, the window area should be at least 10 % of the room's gross internal area (Direktoratet for byggkvalitet, 2011). Other requirements, such as fire safety or installation of lifts etc. was considered irrelevant for the purpose and was therefore not accounted for.

Energy requirements

The energy section of the regulations contains energy efficiency requirements, both for the whole building and building components, and requirements regarding the energy supply of buildings.

The building's maximum total net energy demand is dependent on the building category and are calculated according to NS 3031:2014 "Calculation of energy performance of buildings – Method and data" (Norwegian Building Authority, 2017). The maximum total net energy demand for each building category can be found in the appendix.

In addition to the total net energy requirements, all buildings are obliged to fulfill the minimum energy efficiency requirements for outdoor walls, roof, floor against the ground, windows and doors, and air infiltration rate. It is not necessary for each component to fulfill the requirement, it is satisfactory if the average of all the components in each category fulfills the requirement (Norwegian Building Authority, 2017). The minimum requirements for each component according to TEK17 can be found in the appendix.

For the energy supply, it is not allowed to install heating solutions driven by fossil fuel. Additionally, buildings over 1000 m^2 should facilitate low temperature heating solutions and must have energy flexible heating systems, meaning that it should have at least two energy sources (Norwegian Building Authority, 2017).

NS 3701:2012 "Criteria for passive houses and low energy buildings – Non-residential buildings"

The passive house definition, a certification scheme for building products and buildings, was introduced by the German Passivhaus Institute and have successfully penetrated large markets such as Germany, Austria and other European countries. Given the difference in climate and construction solutions, the definition has been modified to fit the Norwegian conditions, resulting in two standards, NS 3700:2012 for residential buildings and NS 3701:2012 for non-residential buildings. The standards contain Norwegian definitions of passive houses and low energy buildings, with energy requirements, calculation criteria, and criteria for certification and classing of passive houses and low energy buildings. The requirements and criteria were supported academical by a SINTEF Byggforsk study and report conducted for this purpose (Dokka, 2012) (Standard Norge, 2012).

As mentioned the NS 3701:2012 standard specifies the requirements for two different energy efficiency levels, passive house requirements being the strictest. The standard contains minimum requirements of building components, air leakage, heat loss, heating demand, cooling demand, and energy use from lighting and technical equipment (Standard Norge, 2012).

Cooling demand requirements

The maximum calculated net specific cooling demand is given by the equation underneath, based on the dimensioning outdoor temperature (DUTs) of where the building is placed and the cooling coefficient of the selected building standard and category. The DUTs is the temperature that is not exceeded more than 50 hours per year on average at the building location, while the cooling coefficient for each building category is fixed If the DUTs is 20° Celsius or lower, the maximum calculated net specific cooling demand is 0 (Standard Norge, 2012). The cooling coefficient for each building category can be found in the appendix.

$$Maximum\ net\ specific\ cooling\ demand\ (kWh/(m^2 \cdot year)) = \beta(DUTs - 20)$$

Heating demand requirements

The maximum calculated net specific heating demand allowed is decided based on several parameters, and the calculation method varies depending on the building's gross area and the annual mean temperature at the building's location. With the annual mean temperature in Aalesund being above 6.3° Celsius and the projected building's gross area larger than 1000 square meters, the maximum calculated net specific heating demand can be found directly from the table in the appendix (Standard Norge, 2012).

Contradictions between TEK 17 and the NS 3701:2012 standard

As stated previously, TEK 17 is the regulation on technical requirement for construction works in Norway, meaning that it sets the minimum requirements that a building must fulfill. The passive house and low energy building definitions on the other hand, are designed to create high quality buildings with very low energy demand. However, different input values in the PH/LEB calculation compared to TEK 17 calculation can make the TEK 17 regulations stricter, and in some cases stricter than the PH/LEB requirements, making the maximum total net energy requirement in TEK 17 the deciding factor. The reason for this is that the TEK 17 calculations require the use of normative values for climate, technical equipment and lighting, while the PH/LEB definitions use local climate and more efficient technical equipment and lighting.

The normative climate used to calculate the total net energy demand of the building is more extreme than for example the west-coast climate in Aalesund. By calculating with hotter summers and colder winters, the calculations will result in an artificially high heating and cooling demand compared to the PH/LEB calculations that uses local climate.

The PH and LEB standard presumes that technical equipment and lighting are efficient and controlled by a demand control system, resulting in a low internal heat gain. The TEK 17 regulations on the other hand, uses fixed normative values for energy use and heat gain from NS 3031:2014 for the technical equipment and lighting. However, TEK 17 allows reducing the lighting value if a demand control system is documented. These differences, especially the technical equipment, will create an artificially high energy demand in the TEK 17 calculation if the technical equipment and lighting follows the PH and LEB regulations but must be calculated with normative values from NS 3031:2014. The values of both the normative values for TEK 17 and the values for PH and LEB can be found in the appendix.

3.2 Reference buildings

The thought was to project three versions of a 4000m² office building, following the criteria for passive houses, low energy buildings and the minimum requirements of TEK17 respectively. However, simulations showed that he TEK 17 ended up being stricter than the low energy building criteria. The reason behind this is that the TEK 17 requirements are calculated with normalized (Oslo) climate rather than the local climate (Aalesund). It was therefore projected only two versions (PH and LEB) and an additional version where the requirements of the LEB was doubled.

3.2.1 Building layout and general specifications

The building is constructed over four floors, each containing multiple offices, two bathrooms, a conference room and a break room. The 1st floor contains an entrance/reception and a cafeteria in the break room, while the other floors have three additional offices instead. Both versions of the building have the same amount and size of windows and doors except for the 20.25m^3 offices where the size of the windows varies between the versions. The U-value and height of indoor walls are similar for both. The general building specifications for both versions and the building layout can be found in the appendix.

3.2.2 Attributes and performance of the projected reference Passive House and Low Energy Building

As mentioned the Passive House and Low Energy Building were projected to follow the minimum requirements of the PH/LEB standard and TEK 17. The ventilation heat recovery efficiency was set at 82% for both cases as the AHU unit used had this efficiency. To optimize the building towards the maximum cooling demand set by the PH/LEB standard, the attributes of the different building parts such as U-Values were modified within the standard and TEK17 requirements. This was done with local heating and cooling in each room, and ventilation restricted to minimum 7 $\mathrm{m}^3/\mathrm{m}^2h$ and maximum 12 $\mathrm{m}^3/\mathrm{m}^2h$ during office hours and minimum 2 $\mathrm{m}^3/\mathrm{m}^2h$ outside office hours for all rooms. The attributes and performance of the buildings can be found in the appendix along with the power requirement of the heating and cooling equipment for each building.

Using the projected reference building, customized simulations were made to obtain outdoor temperature, demand and fan power data in hourly resolution for solutions using regular AHU and the desiccant solution with integrated air handling. As both the solutions with regular AHU and the desiccant solution cool the room by supplying cooled air, the cooling method was switched to air cooling batteries in the SIMIEN model. The temperature of the inlet air was also set to 17 degrees Celsius for both cases. Even though the SIMIEN simulation results show that cooling is required even at 17 degrees and lower, this demand is considered to be covered by free cooling and is removed from the demand in the performance and cost calculation model. The difference between the simulations, is that the ventilation heat recovery efficiency is higher for the desiccant cooling solution and SFP factor is slightly higher.

3.3 Cost of electricity and district heat

To perform a cost analysis of the different heating and cooling solutions, the cost of electricity and district heat are key parameters. As the electricity price is hard to forecast over decades, the prices were considered to follow the yearly inflation and otherwise remain constant for all years of the analysis.

The electricity price in Norway is built up by fixed, peak power and consumption based costs. The district heat in Aalesund is currently being priced based on the spot price on Nord Pool and has the same fixed, peak power and consumption based costs, except for the profit add on charged by the electricity trading company (Seljelid, 2018).

3.3.1 Consumption based electricity and district heat costs

The consumption based cost formula for electricity is given in equation (1) and consists of the spot price, electricity tax and grid tariff. The spot price is the price payed in the Nord Pool market, which is the Europe's leading power market and where most of the electricity going to regular consumers in Norway is traded (Pool). After the trading companies have bought the electricity on Nord Pool they add a profit margin when selling it to the end user (Fjordkraft). The electricity tax is a tax set by the government, which the grid company collects along with their own grid tariff.

(1) $\boldsymbol{El.price}\ (\boldsymbol{øre/kWh}) = Spot\ price + profit\ add\ on + electricity\ tax + grid\ tariff\ (summer/winter)$

(2) $\boldsymbol{DH\ price}\ (\boldsymbol{øre/kWh}) = Spot\ price + electricity\ tax + grid\ tariff\ (summer/winter)$

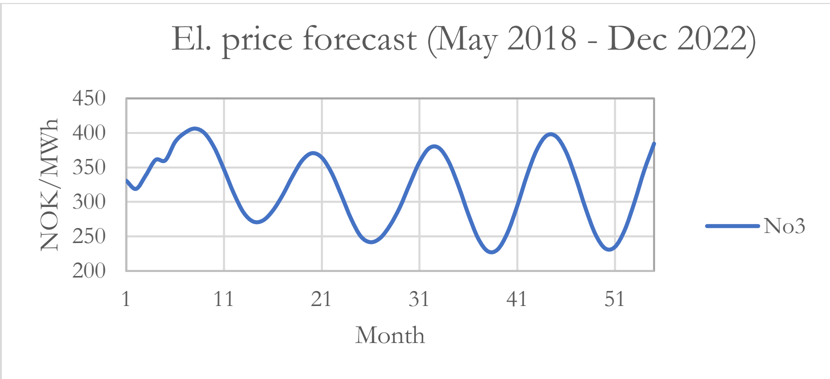

Figure 3.2: Electricity price forecast from the software "Brady Energy Trading and Risk Management".

The spot prices are based on a four-year monthly spot price forecast from the software "Brady Energy Trading and Risk Management", which is a "trading and risk management software for global commodity markets" (network, Cambridge). From figure 3.2 it can be seen that the electricity traded in the future is traded at a higher price during the winter and lower during the summer, this is assumed to be because of discount for uncertainty given by the traders. The spot price used is the average spot price in each month during these forecasted years, and is given in figure 3.3.

Figure 3.3: The avarage spot price for each month during the forecasted years.

The electricity tax, grid tariff and add on used in this thesis are shown in the table below. The tax and tariff are 2018 prices set by the government and the local grid company Morenett, while the profit add on used is the one Fjordkraft uses in their low-price spot deal (Mørenett) (Fjordkraft).

Table 3.1: Profit add on, electricity tax and grid tariff.

Profit add on	***Electricity tax***	***Grid tariff (summer)***	***Grid tariff (winter)***
%	$Øre/kWh$	$Øre/kWh$	$Øre/kWh$
3.5	16.58	4.5	7

After adding the electricity tax, grid tariff and profit margin (electricity) to the electricity and district heat price, the district heat price remains slightly lower than the electricity as expected. This can be seen in the graph underneath where the final monthly prices are plotted.

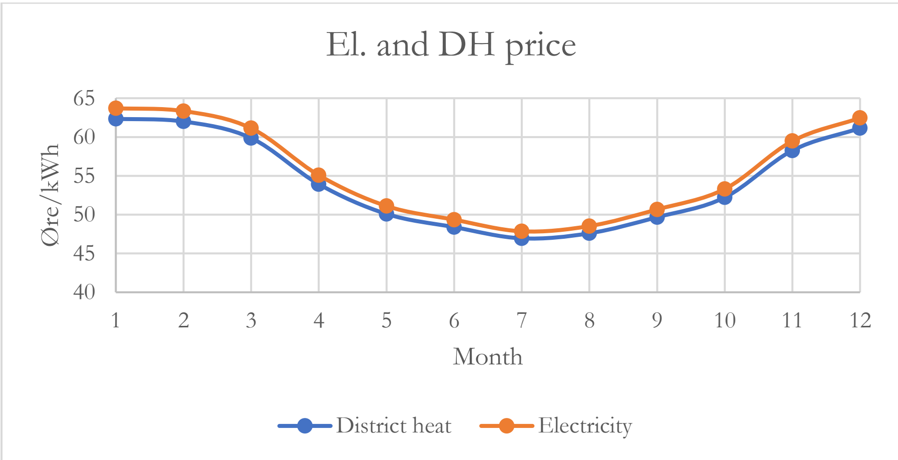

Figure 3.4: Electricity and DH price after adding profit, tax and grid tariff.

3.3.2 Power tariffs and fixed costs

As mentioned, the DH and electricity costs include a fixed cost and power tariffs. While the fixed cost is a fixed sum per year, the power tariff is charged monthly based on the peak power in the given month. The 2018 prices, which are the ones used for this thesis, are similar for both and presented in the table below.

Table 3.2: Fixed cost and power tariffs for 2018 (Mørenett).

Fixed cost	***Power tariff (summer)***	***Power tariff (winter)***
$\boldsymbol{NOK/year}$	$Øre/kW$ (max kW $/month$)	$Øre/kW$ (max kW $/month$)
8800	4.5	7

3.4 Inflation and interest rate

Another key parameter when performing an economic analysis ranging over several years is the inflation rate. As this too is hard to estimate, a constant future inflation rate was forecasted by taking the average of the yearly average CPI inflation in Norway for the past twenty years (1998 – 2017) (Inflation.eu). This resulted in a constant yearly inflation rate of 2.08%.

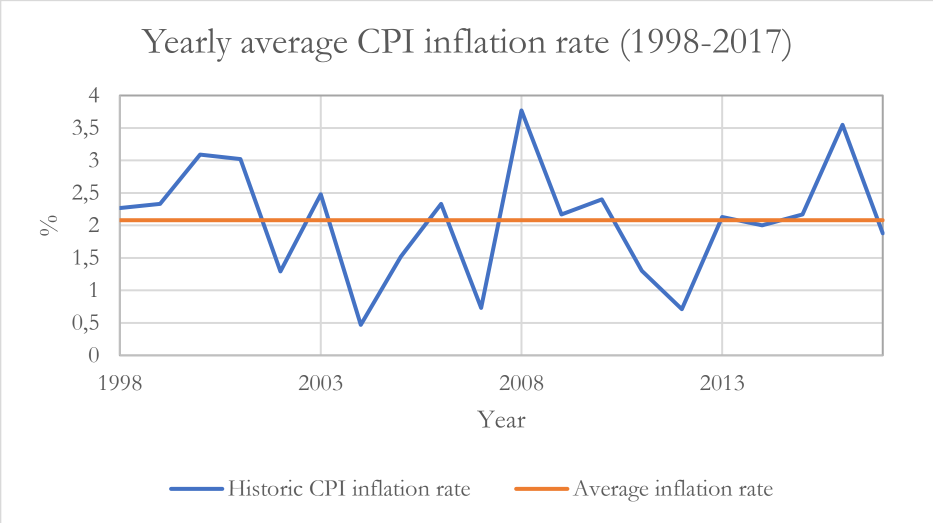

Figure 3.5: Yearly average CPI inflation in Norway for the past twenty years and the average for all twenty years (Inflation.eu).

The interest rate used in the calculations is 2.5%. This is based on SSB's reported interest rates in banks and mortgage companies. They reported an average interest rate of 2.47% in March 2018, and it was decided to round it up to 2.5% to simplify the use of the calculations as the interest rate obtained may vary between projects (Statistics Norway, 2018).

3.5 Heating and Cooling equipment

The heating and cooling solutions considered in the techno-economic evaluation, are the two heat driven cooling solutions and three electrical cooling solutions. One is an electric driven air cooled liquid chiller where district heat is used for heating, while the two others consists of a reverse heat pump assisted by either district heat or an electric boiler during winter peak hours. The air handling unit is similar for all solutions except for the desiccant cooling, which has integrated air handling unit in the cooling machine. Based on offers from distributors one set of heating and cooling equipment for each solution is selected for each building case.

Tables containing the investment cost, service cost and lifetime of the AHU, accumulator tank and the equipment related to each cooling and heating solution can be found in the appendix. As the investment cost of all the equipment are based on offers or estimates by distributors, it is likely to be subject to a profit add by an energy consultant or entrepreneur. A 25% profit add on is therefore included in the investment cost of all the equipment.

3.5.1 Air handling unit

Except for the desiccant cooling machine, which has an integrated air handling unit, all cooling solution requires a AHU to distribute the cooling. The air handling unit used in all solutions is a VEX 4070 unit from Exhausto. According to simulations received from the distributor, the unit can deliver 12000 m3/h of air against 200 Pa of external pressure, with an average heat recovery of 82.4% and an SFP factor of 1.49 kJ/m3. Hence the maximum capacity for the PH and LEB is set to 48000 m3/h, and twice as much in the 2x LEB case, four and eight ventilation units has to be installed in the respective cases. Because the SIMIEN simulations do not allow decimals to be inserted, the heat recovery factor is set to 82% (Andersen, 2018).

3.5.2 Accumulator tank

As all cooling solutions except for the desiccant cooling are producing chilled water, they require an accumulator tank. The tanks used is a VKG 1500 for all chilled water producing solutions in the PH and LEB, and a VKG 3000 for the ones in the 2x LEB case. The tanks can house 1500 and 3000 liters of water in the range of -15 to 60 degrees Celsius. The lifetime of the tanks is estimated to 15 years (A. R. Engen, 2018).

3.5.3 Heat pump

In the three heat pump solutions, two different brands are selected. The two smallest heat pumps are from Aermec, while the largest is a Daiken heat pump. Each pump is described in its own section.

Passive house

For the passive house case, Aermec's NRL 500 HLJ reverse heat pump is selected as the main cooling and heating source, accompanied by an EP 112 electric boiler or district heat during winter time peak hours.

The pump is a three-step heat pump, using R-410A as the refrigerant and it is assumed to be 30% glycol in the water. Performance data is collected from the machines technical manual, where full load input and output values are presented for some temperatures. For the temperatures in between, the performance is estimated by interpolation between known values. The performance at the part load steps is calculated by reducing the full load performance by the percentage reduction input and output for each step, which is also found in the technical manual (Aermec).

It can be seen from the figures below that the coefficient of performance is increased/decreased with changing outdoor temperature. Running the machine at part load would also shift up the temperature dependent performance curve. For the electrical boiler and district heat the COP is considered to be 1.

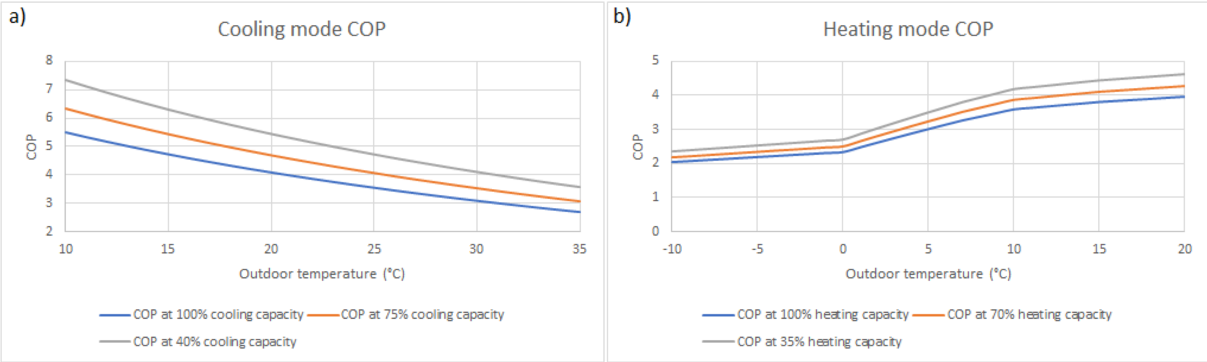

Figure 3.6: a) NRL 500 HLJ performance curve in cooling mode. b) NRL 500 HLJ performance curve in heating mode 40/35.

The costs and lifetime of the machinery is presented in the appendix. The investment costs are based on an offer from a distributor, while the service costs are an estimate of a service deal in the Oslo area (A. R. Engen, 2018).

Low energy building

For the LEB case, an Aermec NRL 600 HLJ reverse heat pump is selected as the main cooling and heating source, accompanied by an EP 180 electric boiler or district heat during peak heating hours.

The pump is in the same series as the pump used in the passive house case, with the main difference being that it has four steps and higher cooling and heating capacities. The performance data is collected from the same manual and in the same manner, and it is therefore derived similarly as well. For the electrical boiler and district heat the COP is considered to be 1 (Aermec).

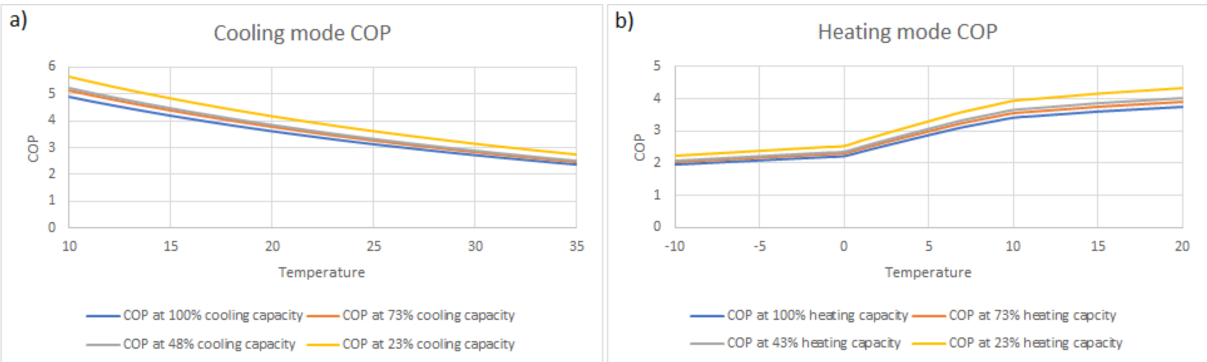

Figure 3.7: a) NRL 600 HLJ performance curve in cooling mode 7/12. b) NRL 600 HLJ performance curve in heating mode 40/35.

The investment costs and service cost estimates are provided by the same distributor as the heat pump solution in the passive house case (A. R. Engen, 2018).

2x Low energy building

The heat pump solution selected for the 2x LEB case is a Daikin EWYQ230F-XL reverse heat pump accompanied by an EP 350 electric boiler or district heat.

The HP is a four-step machine using R-410A as refrigerant and it is assumed to be 30% glycol in the water. Performance estimates for different temperatures are based on input and output values at a few known temperatures, while the unknown data is estimated by interpolation. The output values are also reduced by

5% to account for the glycol in the water on the recommendation of the distributor. Different from the heat pumps in the other cases, the Daikin manual does only provide performance data from 25 Celsius and up in cooling mode. The slope between 10 and 25 Celsius is therefore assumed to be similar to the slope between 25 and 30 Celsius (DAIKIN, 2014) (E. Løberg, 2018).

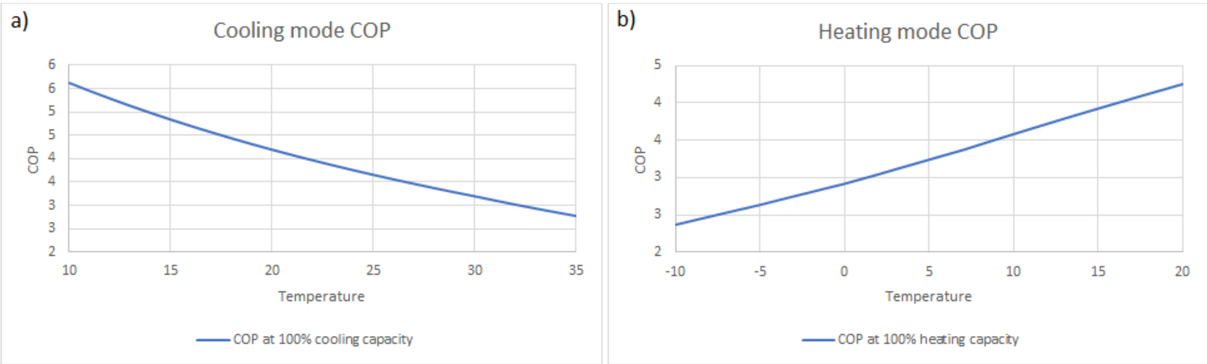

Figure 3.8: a) Daikin EWYQ230F-XL performance curve in cooling mode 7/12. b) Daikin EWYQ230F-XL performance curve in heating mode 40/35.

The investment costs of the machinery are based on an offer from a distributor, while the service cost is an estimate provided by the same distributor (E. Løberg, 2018).

3.5.4 Absorption cooling

The absorption cooling solutions consist of an absorption chiller with a dry cooler working as the heat sink, while the heating demand is met by district heat.

All three absorption chillers used are from Yazaki Nordic and are selected based on cooling capacity required in each of the building cases. A WFC-M100, Yazaki's larges chiller, is selected to meet the cooling demand in the 2x LEB case, while in both the LEB and passive house case a WTC-SC30 is the chiller of choice. These are water fired chillers using water/LiBr as their working pair, meaning that there are no environmentally unfriendly freons (Yazaki) (Yazaki Nordic).

The optimal running state of each chiller, obtaining maximum COP while still meeting the cooling capacity requirements, was found using Yazaki's own calculation model. The model allows the temperature lift and power requirement of the driving heat and heat sink to be found by adjusting their inlet temperature (Yazaki Nordic, 2018).

Table 3.3: The optimal running state for each building case.

	Passive House	LEB	2x LEB	
Chilled water temperature	7/12	7/12	7/12	°C
Driving heat inlet temperature	87/81.3	92/86	80/73.6	°C
Heat sink temperature	35/31	35/31	33.5/29.5	°C

The dry cooler for each chiller is chosen using AIACalc software, where a dry cooler is selected based on required power and input/output temperature of the heat sink at a given out door temperature (DUTs)

(AIA (LU-VE Sweden)). After selecting a dry cooler fulfilling the heat sink requirements at the DUTs, the dry cooler's performance curve was created for declining outdoor temperatures in steps of one degree. This was done by reducing the RPM till the edge where the power requirements of the heat sink were still met.

The absorption chiller's dip in cooling capacity at 23 Celsius originates from the lack of cooling capacity from the heat sink. This is because the dry cooler is dimensioned based on required cooling capacity at DUTs (22.4 Celsius) and at higher temperatures it is unable to provide sufficient cooling.

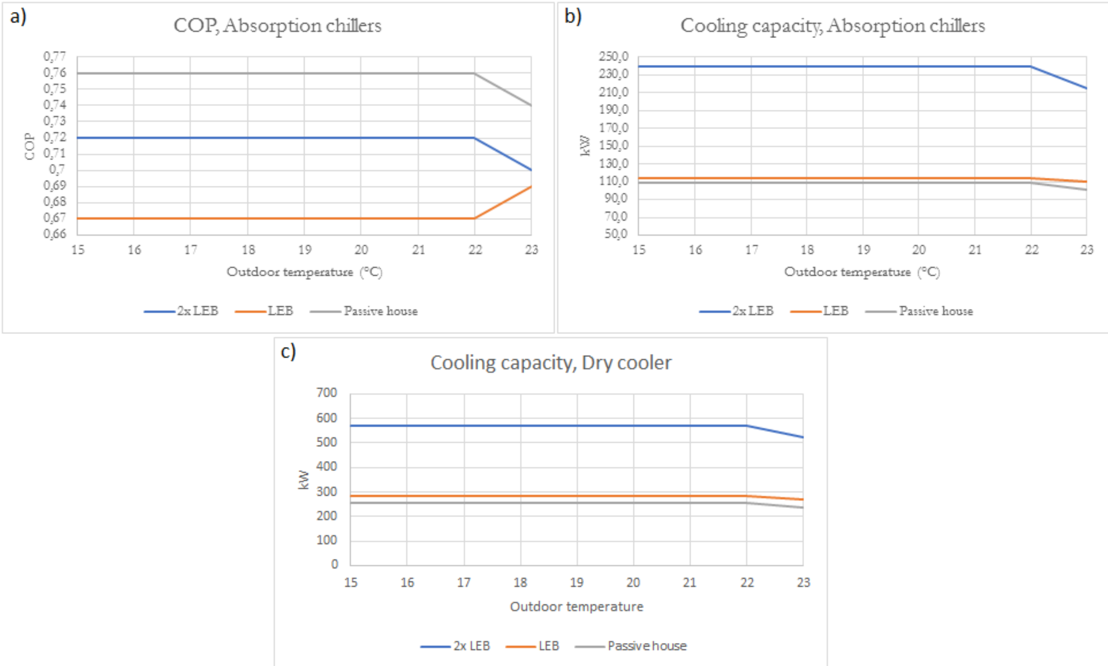

Figure 3.9: a) COP of the absorption chillers at chosen running conditions at different outdoor temperatures. b) Cooling capacity at the same running conditions and temperatures. c) Cooling capacity of the dry cooler at chosen running conditions at different outdoor temperatures.

The chiller investment costs are estimated by the distributor and includes the chiller, transportation, testing, warranty, control system and some unspecified costs (300 euro). The dry cooler investment cost is estimated by the same distributor, but is accounted for separately as the lifetime is under half of the chiller's lifetime. Both investment costs are given in euros and transformed from Euro to NOK using the exchange rate of 9.6 Euro/NOK. This means that the costs are subject to change due to fluctuations in the Euro/NOK exchange rate. For the service cost, the distributor estimates that 750 NOK should be enough to have a service technician looking at the machinery every fifth year. The dry cooler is considered part of the chiller's service. The expected lifetime of the machinery, is 40 years for the chiller and 15 years for the dry cooler. The lifetime of 40 years is an assumption made by the distributor based on existing machinery (A. Nesthorne, 2018)

3.5.5 Electric driven air cooled liquid chiller

The air cooled liquid chiller solutions consist of an electrically driven chiller for cooling, while the heat demand is met by district heat. All three chillers selected are scroll chillers from Carrier's Aquasnap series and are selected based on cooling capacity required in each of the building cases.

The selected chillers are: Aquasnap 30RBS-090C, Aquasnap 30RBS-120C and Aquasnap 30RBP-0220 for the passive house, LEB and 2xLEB. While the two smallest chillers have three load steps, the largest chiller has four steps and a green speed intelligence system. They are all using R410-A as a refrigerant and it is assumed to be 30% glycol in the water (Thermo Control, 2018) (Thermo Control, 2018) (Thermo Control, 2018).

Performance data for each chiller is obtained from performance simulations done by the distributor. The simulations show the cooling capacity and input power at each capacity step for several different outdoor temperatures. The simulations start at 35 degrees Celsius and are performed in steps of 5 degrees, ending at 10 degrees (11 degrees for the two smallest chillers). For the remaining temperatures, values are found by interpolation between the simulated values (Thermo Control, 2018) (Thermo Control, 2018) (Thermo Control, 2018).

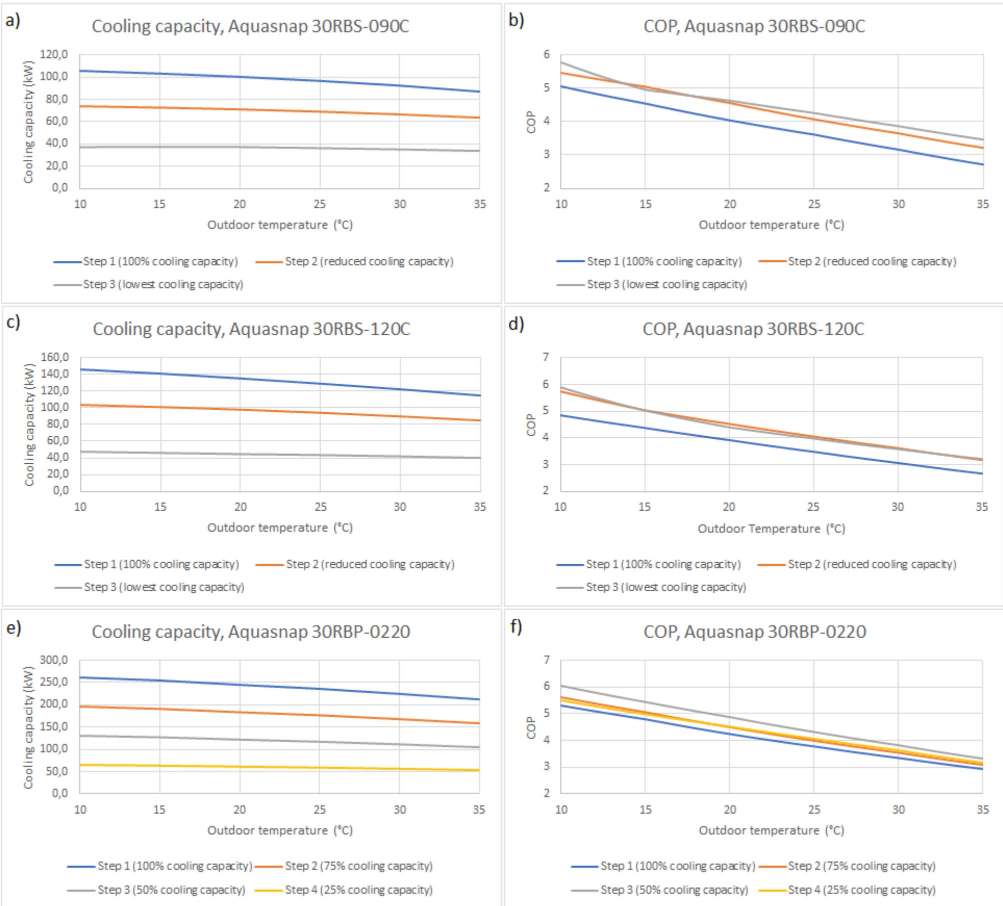

Figure 3.10: ***a)*** *Cooling capacity curves for the electric chiller in the Passive house case.* ***b)*** *COP curves for the electric chiller in the Passive house case.* ***c)*** *Cooling capacity curves for the electric chiller in the LEB case.* ***d)*** *COP curves for the electric chiller in LEB case.* ***e)*** *Cooling capacity curves for the electric chiller in the 2xLEB case.* ***f)*** *COP curves for the electric chiller in the passive house case.*

The investment costs of the chillers are based on an offer from a distributor that used a NOK/Euro exchange rate of 9.6, while the service cost and lifetime is an estimate provided by the same distributor. The offer does not include construction work, plumbing, lift, electrical connection or charges due to loss of refrigerant (T. Lie, 2018).

3.5.6 Desiccant cooling

The desiccant cooling solutions consists of a desiccant cooling machine, and works as an air handling unit with integrated cooling, which also has enhanced heat recovery compared to a regular air handling unit. The heating demand is satisfied by district heat (Anders Granstrand, 2018).

The same desiccant cooling machine is selected for all three cases, with two machines in the 2xLEB case to satisfy the cooling demand. The machine is a DesiCool DCI 13.7 from Munters Europe AB.

As the performance of the machine is dependent of several factors, such as the inlet temperature to the building, outdoor temperature, relative humidity, SFP factor, some assumptions were made.

Since the relative humidity can vary drastically even for the same temperature, which heavily changes the performance and district heat input of the machine, a data set containing temperatures and relative humidity from the local airport between the years 2004 and 2017 were analyzed. By looking into the data, the time spent at each relative humidity (steps of 10%) for each temperature was found (table in appendix). Based on these numbers the distributor performed simulations producing 48000 m3/h of inlet air at a temperature of 17 Celsius for each of the relative humidity intervals at each temperature step. The COP was found for each relative humidity step with time percentage above zero, and this was used to find the average COP for each temperature step, resulting in the performance curve in figure 3.11 (eKlima, 2018) (Anders Granstrand, 2018).

According to the distributor, the SFP factor can fluctuate from barely over 1 to barely over 2 throughout the year, with an estimated average between 1.5 and 1.9. An assumed SFP factor of 1.5 was therefore used in the SIMIEN simulations. The remaining electricity consumption of the unit was neglected, as the distributor considered it to be very much lower than the fan consumption. The heat recovery of the ventilation air is also dependent on the outdoor conditions, and an estimate on average about 90% were used on recommendation for the distributor (Anders Granstrand, 2018).

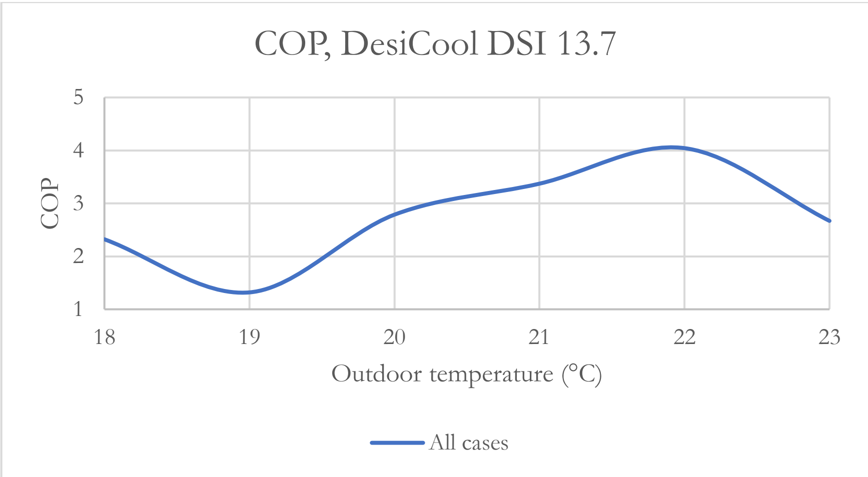

Figure 3.11:. Estimated performance curve of the desiccant cooling machine.

The investment cost was estimated by the distributor to be between 1.5-1.6M SEK, and 1.55M SEK is therefore used as the price from the distributor. The investment cost is calculated using a SEK/NOK exchange rate of 0.9175 (Norges Bank, 2018). The cost includes start up, while assembly of the electric, steering, ventilation and piping is not included. The distributor estimates the same lifetime and slightly higher service costs than a regular air handling unit, while the cooling part of the machine does not require any service. The service cost is therefore assumed to be 15% higher than the air handling unit in section 3.4.1. (A. Grandstrand, 2018).

3.6 Performance and cost calculation model

To perform the techno-economic evaluation and benchmarking the heat driven cooling solutions against regular heat pumps and electric chillers, a performance and cost calculation model was built in Excel. The model was built using 15 separate sheets that can be classified as input data sheets, performance/cost calculation sheets and a final sheet containing levelized cost calculations for all solutions, and input fields for some economic variables.

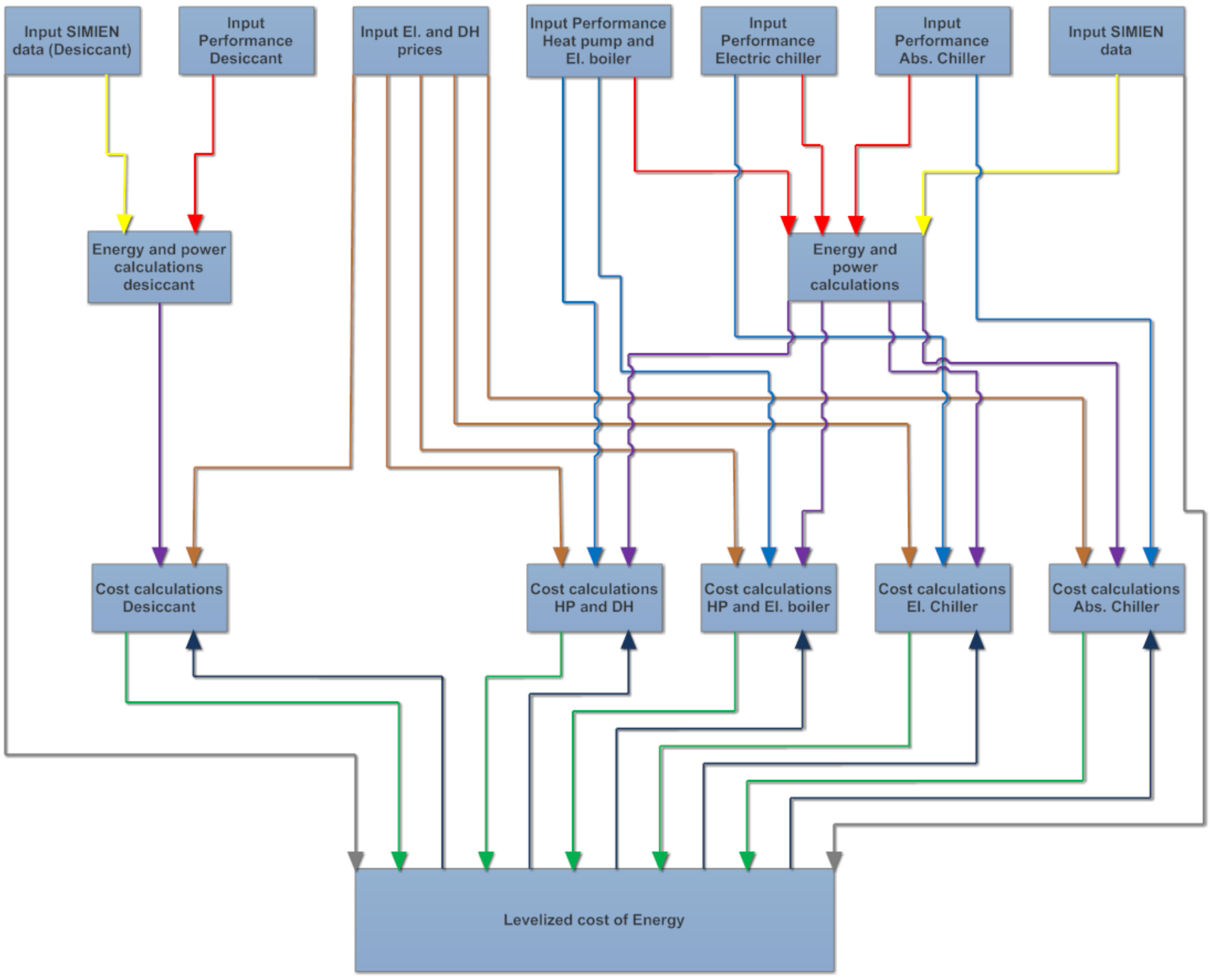

Figure 3.12: Build-up and information flow for the performance and cost calculation model.

3.6.1 Information flow between sheets

The sheets in the model exchange information to perform the calculations for further or final use. The arrows in figure 3.12 show the data flow between the sheets in the calculation model. They are divided by color based on what kind of information is sent.

The yellow arrows illustrate the data flow from the Input SIMIEN sheet to the Energy and power calculation sheet. Fan power, outdoor temperature, cooling requirement and heating requirement in hourly resolution throughout the year is collected by the Energy and power calculation sheets.

The red arrows illustrate the data flow from the input performance sheets for each technology to the Energy and power calculation sheets. The Energy and power calculation sheets collects performance data of the equipment to perform the energy and power calculations.

The blue arrows illustrate the data flow from the input performance sheets for each technology to the Cost calculation sheets. The Cost calculation sheets collect the costs and lifetimes of the equipment for each solution and use it to calculate yearly and lifetime costs.

The brown arrows illustrate the information flow from the Input electricity and district heat price sheet to the Cost calculation sheets. The Cost calculation sheet uses the electricity and district heat prices to calculate monthly, yearly and lifetime cost of electricity and district heat.

The purple arrows illustrate the information flow from the Energy and power calculation sheets to the Cost calculation sheets. The Cost calculation sheet uses monthly energy consumption and peak power from the Energy and power calculation sheets to calculate monthly, yearly and lifetime costs of electricity and district heat.

The dark blue arrows illustrate the information flow from the Levelized Cost of Energy sheet to the Cost calculation sheets. The Cost calculation sheets use economical inputs from the Levelized Cost of Energy sheet to perform the lifetime cost calculations.

The green arrows illustrate the information flow from the Cost calculation sheets to the Levelized Cost of Energy sheet. The lifetime costs from the Cost calculation sheet are used to calculate the LCOE for each solution.

The grey arrows illustrate the data flow from the Input SIMIEN sheets to the Levelized Cost of Energy sheet. The annual cooling and heating requirements from the Input SIMIEN sheets are used to perform the LCOE calculations.

3.6.2 Input Sheets

The calculations in the model are based on data from three kinds of input data sheets and some economical parameters in the Levelized Cost of Energy sheet. The input sheets can be categorized into three subdivisions, the SIMIEN data, energy prices and performance sheets.

The data from the full year simulation in SIMIEN, such as outdoor temperature, the cooling and heating requirements and fan power for each timestep are in the input SIMIEN sheets. Because the desiccant solution has an integrated ventilation system with a higher heat recovery than the air handling unit used in the other solutions, a separate SIMIEN simulation was performed. This is the reason behind the separate input SIMIEN sheet for the desiccant cooling solution.

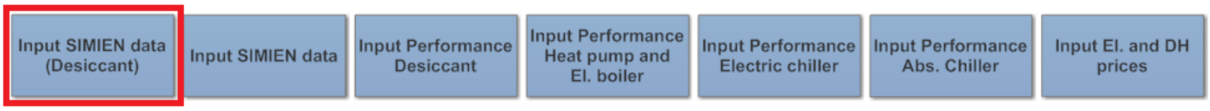

Figure 3.13: A separate input SIMIEN sheet for the desiccant cooling solution.

The "Input performance" sheets contain the performance of the cooling and heating equipment at different outdoor temperatures and at different load stages. It also contains the costs and lifetime of both the cooling and heating, and the ventilation equipment. In cases where district heat is used as the heat source, the equipment cost for heating is considered to be zero and the amount district heat delivered for space heating is considered to be equal to the demand of heat.

3.6.3 Energy and power calculations

In the energy and power calculation sheets, the power input and electricity consumption for each piece of equipment is calculated in an hourly resolution. This is done using the outdoor temperature and cooling/heating demand from the SIMIEN simulation, and the performance data for the equipment. The electricity consumption of the air handling unit is also collected from the SIMIEN simulation and converted from Watt to kilowatt. Additionally, the monthly peak electricity and/or district heat power is calculated for each solution, this also includes the electric power used for the air handling unit.

For all cases and solutions, the outdoor temperature from the input SIMIEN sheet is rounded to the nearest integer using the "MROUND" function in excel. This is for the temperature to be compatible with the performance data, which is also given in a temperature resolution of 1. The cooling demand for outdoor temperatures below 18 degrees (after rounding) is set to zero. This is because the cooling machines are set to produce inlet air of 17 degrees, meaning that at outdoor temperatures at 17 degrees or lower no cooling power is used to cool the supply air.

Heat pump and electric boiler/district heat solution (hourly values)

To find the monthly energy consumption and input peak power for the heat pump and electric boiler/district heat solutions, several part calculations had to be made. The calculations are similar for both the heating and cooling, exception for the boiler/district heat peak load calculations, which is performed solely for heating. Each one of the heating calculations are described below, a short description of the difference for cooling calculations will be given at the end of each explanation.

Power input heat pump

The input power is found using the IF, INDEX and MATCH functions in excel. The purpose of the INDEX function is to find the row and column within the array selected. The data is set up so that the temperatures are placed in a vertical column and the corresponding heating capacity steps are placed in the horizontal row. An array is selected containing the performance data for the heat pump in heating mode, for all steps in the temperature range 20°C to -10°C.

To select the correct row that corresponds to the outdoor temperature at the given timestep, the following code is used (only for explanation, not the actual code):

```
IF outdoor temperature <= 20;
True: MATCH (outdoor temperature; range(row with 20 Celsius to row with -10);0)
False: Cell containing 20 Celsius is selected)
```

The IF function ensures that the match function is used only for temperatures below 20°C, if false, the heating capacities for 20°C are used. The MATCH function finds the row that corresponds to the outdoor temperature, by starting at 20°C and scanning downwards until the outdoor temperature matches the row value. The code does not take into consideration temperatures lower than -10°C, simply because it is not in the SIMIEN data.

To find the column (step), the following code is used (only for explanation, not the actual code):

```
IF outdoor temperature >= 20 Celsius AND heating requirement <= heating
capacity of the lowest step: (True: Column with the input power corresponding
to the lowest heating capacity is selected. False: Next IF)

IF outdoor temperature >= 20 Celsius AND heating requirement > heating capacity
of the lowest step: (True: (MATCH (First heating capacity to be >heating demand
(range: lowest to highest heating capacity)) The MATCH function provides an
index number which locates the correct input power. False: Next IF)

IF outdoor temperature = 19 Celsius AND heating requirement <= heating capacity
of the lowest step: (True: Column with the input power corresponding to the
lowest heating capacity is selected. False: Next IF)

IF outdoor temperature = 19 Celsius AND heating requirement > heating capacity
of the lowest step: (True: (MATCH (First heating capacity to be >heating demand
(range: lowest to highest heating capacity)) The MATCH function provides an
index number which locates the correct input power. False: Next IF)

.

Repeats: 20 -> -10 degrees.

.

IF outdoor temperature = -10 Celsius AND heating requirement <= heating
capacity of the lowest step: (True: column with the lowest heating capacity is
selected. False: Next IF)

IF outdoor temperature = -10 Celsius AND heating requirement > heating capacity
of the lowest step: (True: (MATCH (First heating capacity to be >heating demand
(range: lowest to highest heating capacity)) The MATCH function provides an
index number which locates the correct input power. False: Next IF)
```

The IF and AND functions determines if the row corresponds to the outdoor temperature and if the heating demand is higher or lower than the heating capacity at the lowest step at that temperature. If the heating demand is lower than the heating capacity at the lowest step for the determined outdoor temperature, the lowest step is selected. If the demand exceeds the capacity, the MATCH function is activated. The MATCH function locates the cell which contains the highest heating capacity that is exceeded by the heating demand. This means that it selects a step that do not fulfill the heating demand, which is why the +1 is added to the step by one. This way the lowest step that meets the heating demand is selected. In cases where the heating demand exceeds the heating capacity the highest capacity step is selected.

The input power calculations in cooling mode is performed in the exact same manner, but in the range of 10°C to 35°C.

Electricity consumption heat pump

Since the heat pumps only have three or four steps, the heating or cooling capacity is unlikely to match the demand. The pump is therefore considered to run at the selected step just long enough for the demand to be met and then shut off until the next timestep. This means that the electricity consumption is not the same as the input power. The electricity consumption of the heat pump is calculated using the following formula:

$$Electricity\ consumption = Heating(cooling)\ demand/COP$$

While the heating and cooling demand are collected directly from the SIMIEN sheet as mentioned, the COP of the heat pump at each timestep is found using the same method as the input power. The only difference is that instead of selecting the input power that corresponds to the heating capacity, the corresponding COP (heating capacity/input power) is selected.

Electric boiler/District heat power and consumption

As mentioned, an electric boiler or district heat is used during heating peak hours when the heat pump is not able to meet all of the heating demand by itself. The electricity/district heat consumption and power are given by the formula:

$$El. boiler/DH\ consumption\ and\ power\ = \ Heating\ demand - Heating\ capacity$$

To figure out if the heating demand is larger than the heating capacity the method used for the input power is used, but instead of selecting the input power the heating capacity itself is selected. The boiler/DH consumption and power are then found using the following code (only for explanation, not the actual code):

```
IF Heating demand > Heating Capacity (Heat pump);
True:  El.boiler/ DH consumption and power = Heating demand - Heating capacity
False: El.boiler/ DH consumption and power = 0
```

Electric chiller and district heat solution (hourly values)

The calculations for the electric chiller is performed in the same manner as the calculation for the heat pump in cooling mode (check previous section). The temperature range is also similar (10°C to 35°C). For the heating calculations, the heating demand and the district heat consumption and power are considered to be 1 to 1.

Absorption cooler and district heat solution (hourly values)

The absorption cooler has only one step and is considered to regulate the cooling by shutting of when the cooling demand is met and then start up again to meet the next cooling demand. In addition to the cooler itself, the electricity use of the dry cooler also had to be considered. The performance of both the absorption cooler and dry cooler has been found for the range 15 to 23°C. The heating is supplied by district heat and is considered to be 1 to 1 with the heating demand.

Because the absorption cooler only has one step, the code for the power and consumption calculations are less complex than the heat pump and electric chiller solutions. However, the consumption of both district heat and electricity, and the dry cooler leads to more calculations.

Absorption cooler input power (District heat)

The input power is found using the IF, INDEX and MATCH functions in excel. The data is set up so that the temperatures are placed in a vertical column and the corresponding input power is placed in the horizontal row. An array is selected containing the input power for each temperature in the range 18°C to 23°C.

To select the correct row that corresponds to the outdoor temperature at the given timestep, the following code were used (only for explanation, not the actual code):

```
IF 24 > outdoor temperature > 17;
True: (MATCH (outdoor temperature; range(row with 18 Celsius to row with
23);0)) The MATCH function provides an index number which locates the correct
input power.
False: Input power = 0)
```

The IF function allows timesteps with temperatures within the range to be further processed. The MATCH function is then used to find the correct index number to locate the input power for the given outdoor temperature.

Absorption cooler consumption of district heat

As mentioned the absorption cooler is considered to run until the cooling demand has been met, and then shut down until the timestep with cooling demand. This means that the consumption of district heat is different from the input power. The consumption is found using the following formula:

$$Abs. cooler\ DH\ consumption = Cooling\ demand\ /\ COP$$

The COP is found using the same method as the input power, but instead of locating the input power the array is changed so that the INDEX function locates the COP instead.

Absorption cooler electricity consumption and power

The electricity use of the absorption chiller is considered to follow the running time of the absorption chiller, while the electricity peak power is the same whether it is running the whole hour or 1 second.

The model calculates the power and consumption in the following manner:

```
IF Cooling demand > 0;
True: Electricity power = X (value changes for different absorption coolers).
False: Input power = 0
```

```
IF Cooling demand > 0;
True: Electricity consumption = Electrical power * Cooling demand / Cooling capacity
False: Electricity consumption = 0
```

Dry cooler electricity consumption and power

The electrical power of the dry cooler is found in the same manner as the input district heat power of the absorption cooler, the only difference being that another array is selected, meaning that the index function locates the dry cooler's input power instead of the input power of the absorption cooler. The electric consumption of the dry cooler is following the electric power in the same manner it does for the absorption cooler.

Desiccant cooling machine and district heat solution (hourly values)

Unlike the other solutions which has separate air handling units, the desiccant cooling machine has an integrated air handling unit so that it provides both the cooling and the fan power. The performance data explained in section 3.6.5 is used in the model along with the cooling requirement obtained in the SIMIEN model. The district heat consumption of the machine is based on an average COP when delivering inlet air at the given outdoor temperature. The fan power, which is the electricity input and power consumption of the desiccant machine, is collected directly from the SIMIEN data after inserting the SFP factor in the SIMIEN model. The district heat consumption and power is considered to be 1 to 1 with the heating demand.

District heat input power and consumption of the desiccant cooling machine

The input power and consumption of district heat is found using the IF, INDEX and MATCH functions in Excel. The data is set up so that the temperatures are placed in a vertical column and the corresponding input power is placed in the horizontal row. An array is selected containing the input power for each temperature in the range 18°C to 23°C.

To select the correct row that corresponds to the outdoor temperature at the given timestep, the following code was used (only for explanation, not the actual code):

```
IF 24 > outdoor temperature > 18;
True: (MATCH (outdoor temperature; range(row with 15 Celsius to row with
23);0)) The MATCH function provides an index number which locates the correct
input power.
False: Input power = 0
```

The IF function allows timesteps with temperatures within the range to be further processed. The MATCH function is then used to find the correct index number to locate the input power for the given outdoor temperature.

Monthly peak power loads for the different solutions

The monthly peak power loads for each solution are based on the hourly input power values found for each piece of equipment in the solution. As the electricity and district heat loads in some of the solutions are built up by several sources, the hourly power values from each source are summed to create the total load of electricity or district heat in the given timestep. After finding the total load for each timestep, the peak load for each month was found using the MAX function in excel for all timesteps of the month. Figure 3.14 shows how the hourly value parameters are distributed and used for peak power calculations in the absorption cooling solutions. The figures for the other solutions can be found in the appendix.

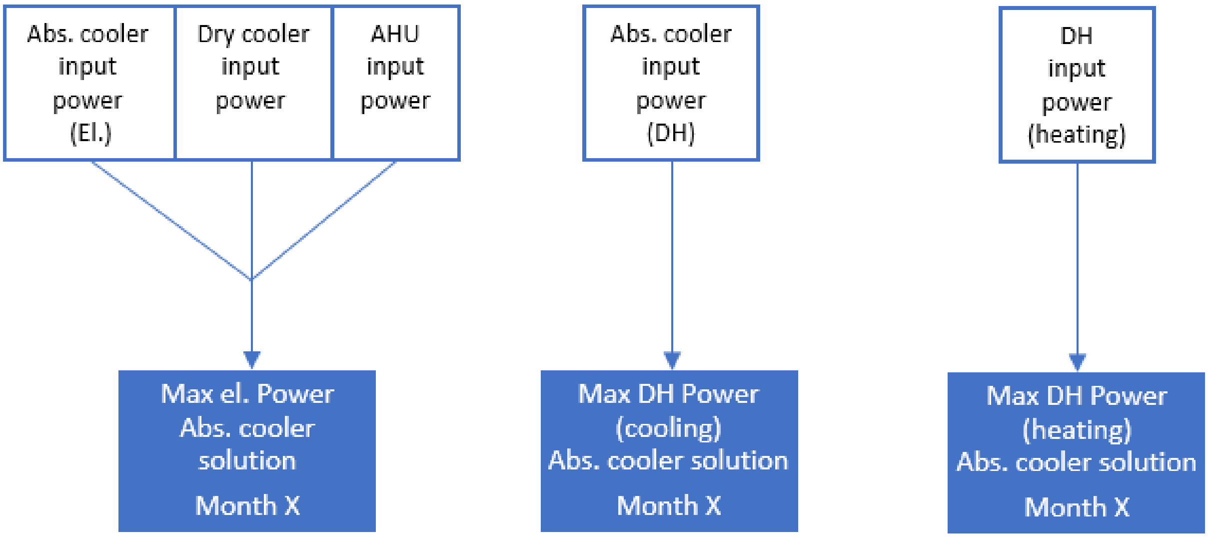

Figure 3.14: How the hourly value parameters are distributed and used for peak power calculations in the absorption cooling solution.

Heat pump and electric boiler solution

For the heat pump and electric boiler solution only the peak power of electricity is found, since there is no use of district heat. The electricity peak power is built up by the heat pump's input power in both heating and cooling mode, the electric boiler input power and the input power of the air handling unit.

Heat pump and district heat solution

Since the heat pump and district heat solution replace the electric boiler from the other HP solution with district heat, the electric input power of the electric boiler is switched to district heat power.

Electric chiller solution

For the electric chiller and district heat solution the electric power is built up by the electric chiller's and the air handling unit's input power, while the district heat power is solely from the district heat used for heating.

Absorption cooling solution

For the Absorption cooling solution, the electric power load is built up by the absorption cooler, dry cooler and air handing unit's input power, while the district heat power is separated into two loads, one for the cooling and one for heating. The reason behind the segregation of the district heat power is to be able to put different prices on the district heat power used for heating and cooling in the cost calculations.

Desiccant cooling solution

As for the Absorption cooling solution, the district heat power is separated into power used for cooling and power used for heating. The electric power for the Desiccant cooling solution consists solely of the electric power used by the desiccant machine.

3.6.4 Cost calculations

All the costs related to each solution are calculated throughout the cooling equipment's lifetime. These calculations are performed in the "Cost Calculation" sheets with the same method used for all solutions. The cost calculated are electricity, district heat, investment costs, interest costs and service costs.

Electricity and district heat costs

Electricity and district heat costs consists of consumption, power and fixed costs. The consumption and power costs are calculated for each month and summed into yearly costs, while the fixed cost is already given on annual basis. These parameters together constitute the yearly electricity/district heat cost.

The consumption cost calculations are performed by summing the hourly consumption of each piece of equipment consuming electricity/district heat for the entire month and multiplying it with the estimated electricity/district heat price of the month.

The power cost calculations are performed by multiplying the peak power loads for each month with the power tariff charged for the month. For the absorption cooling and desiccant cooling solutions, the power tariff on the district heat used for cooling can be tweaked to model how the costs will change if the power tariff is lowered.

It is assumed that prices follow the inflation and that the electricity/district heat consumption and power peaks are similar for all years. The cost of electricity and district heat will therefore stay constant for all years in term of present value.

Equipment costs

The cost of buying and maintaining each piece of equipment is divided into investment cost, interest cost and service cost. These costs are calculated in the same manner for all the equipment throughout the cooling machine's lifetime. Even though the calculation method is similar, there is a slight change to the code for obtaining the lifetime costs of the AHU and dry cooler in the Absorption cooling solution. Since the absorption cooler's lifetime exceeds the lifetime of the dry cooler and AHU, they must be replaced during the lifetime of the Abs. cooler.

Investment cost

As it is assumed that the company that is buying the equipment is financing the investment by loan, the investment costs are divided into yearly installments for half of the lifetime of the equipment. The costs are presented in present value, meaning that the yearly installments are adjusted for inflation.

To find the annual installment for each year for each piece of equipment (except the dry cooler, AHU and accumulator tank in the Abs. cooling solution), the following code is used (only for explanation, not the actual code):

```
IF Year - 1 < Equipment down payment time;
True: Annual installment = Equipment Price / (Equipment's down payment time * (1+Inflation rate)^year)
False: Annual installment = 0
```

$$\text{Annual installment} = \frac{Equipment\ Price}{Equipment's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$

To find the annual installment for each year for the dry cooler, AHU and accumulator tank in the Absorption cooling solution, the following code is used (only for explanation, not the actual code):

```
IF Year - 1 < Equipment down payment time;
True: Annual installment = 
```
$$\frac{Equipment\ Price}{Equipent's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$
```
False: Next IF.
IF Abs. cooler lifetime > Equipment lifetime AND year – 1 = Equipment lifetime;
True: Annual installment = 
```
$$\frac{Equipment\ Price * (1+Inflation\ rate)^{Equipment\ lifetime}}{Equipent's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$
```
False: Next IF.
IF Year > Equipment lifetime AND year – 1 < Equipment lifetime + down payment time AND
Abs. cooler lifetime > Equipment lifetime;
True: Annual installment = 
```
$$\frac{Equipment\ Price * (1+Inflation\ rate)^{Equipment\ lifetime}}{Equipent's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$
```
False: Next IF.
IF Abs. cooler lifetime > Equipment lifetime AND year – 1 = Equipment lifetime * 2;
True: Annual installment = 
```
$$\frac{Equipment\ Price * (1+Inflation\ rate)^{Equipment\ lifetime * 2}}{Equipent's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$
```
False: Next IF.
IF Year > Equipment lifetime * 2 AND year – 1 < Equipment lifetime * 2 + down payment
time AND Abs. cooler lifetime > Equipment lifetime * 2;
True: Annual installment = 
```
$$\frac{Equipment\ Price * (1+Inflation\ rate)^{Equipment\ lifetime * 2}}{Equipent's\ down\ payment\ time * (1+Inflation\ rate)^{year}}$$
```
False: Annual installment = 0.
```

Interest cost

The interest cost of the equipment is calculated in annual resolution, where the remaining amount of loan in the beginning of the year (in real time NOK, not present value) is multiplied with the interest rate.

To find the annual interest cost for each year for each piece of equipment (except the dry cooler, AHU and accumulator tank in the Abs. cooling solution), the following code is used (only for explanation, not the actual code):

```
IF Year = 1;
True: Annual interest cost = 
```
$$\frac{Equipment\ Price * Interest\ rate}{(1+Inflation\ rate)^{year}}$$
```
False: Next IF
IF Year - 1 < Equipment down payment time;
True: Annual interest cost = 
```
$$\frac{\left(Equipment\ Price - \left(\frac{Equipment\ price}{Down\ payment\ time} * (Year-1)\right)\right) * Interest\ rate}{(1+Inflation\ rate)^{year}}$$
```
False: Annual interest cost = 0
```

To find the annual interest cost for each year for the dry cooler, AHU and accumulator tank in the Absorption cooling solution, the following code is used (only for explanation, not the actual code):

IF Year = 1;

True: Annual interest cost = $\frac{Equipment\ Price * Interest\ rate}{(1+Inflation\ rate)^{year}}$

False: Next IF

IF Year - 1 < Equipment down payment time;

True: Annual interest cost = $\frac{\left(Equipment\ Price - \left(\frac{Equipment\ price}{Down\ payment\ time} * (Year-1)\right)\right) * Interest\ rate}{(1+Inflation\ rate)^{year}}$

False: Next IF

IF Year – 1 = Equipment lifetime **AND** Annual installment > 0;

True: Annual interest cost = $\frac{Equipment\ Price * (1+Inflation\ rate)^{Equipment\ lifetime} * Interest\ rate}{(1+Inflation\ rate)^{year}}$

False: Next IF

IF Year > Equipment lifetime **AND** year – 1 < Equipment lifetime + Down payment time **AND** Annual installment >0;

True: Annual interest cost =

$$\frac{\left((Eq.\ Price*(1+Inflation\ rate)^{Eq.\ lifetime}) - \left(\frac{Eq.\ price*(1+Inflation\ rate)^{Eq.\ lifetime}}{Down\ payment\ time} * (Year-1-Eq.\ lifetime)\right)\right) * Interest\ rate}{(1+Inflation\ rate)^{year}}$$

False: Next IF

IF Year – 1 = Equipment lifetime * 2 **AND** Annual installment > 0;

True: Annual interest cost = $\frac{Equipment\ Price * (1+Inflation\ rate)^{Eq.\ lifetime * 2} * Interest\ rate}{(1+Inflation\ rate)^{year}}$

False: Next IF

IF Year > Equipment lifetime * 2 **AND** Year – 1 < Equipment lifetime * 2 + Down payment time **AND** Annual installment >0;

True: Annual interest cost =

$$\frac{\left((Eq.\ Price*(1+Inflation\ rate)^{Eq.\ lifetime * 2}) - \left(\frac{Eq.\ price*(1+Inflation\ rate)^{Eq.\ lifetime * 2}}{Down\ payment\ time} * (Year-1-Eq.\ lifetime * 2)\right)\right) * Interest\ rate}{(1+Inflation\ rate)^{year}}$$

False: Annual interest cost = 0.

Service cost

The service cost related to each piece of equipment is assumed to follow the inflation, and the present value is therefore constant for all years of service. It is also assumed that the first service is required after the first year and the last service is performed one year before the cooling machine is taken out of service.

IF Year < Cooling equipment lifetime;

True: Annual service cost = $Equipment's\ estimated\ annual\ service\ cost$

False: Annual service cost = 0

Lifetime costs

The lifetime costs are found in terms of lifetime cost of electricity and district heat for the whole cooling solution, and the investment, interest and service cost of each piece of equipment during the lifetime of the cooling equipment, and the total lifetime cost for the whole cooling solution.

The lifetime costs related to each piece of equipment and the lifetime electricity and district heat costs are calculated by adding together the calculated annual costs. The total lifetime cost of the cooling solution is then found by summing all the lifetime costs of the solution.

3.6.5 Levelized Cost of Energy

The final sheet calculates the Levelized cost of energy related to each solution. The calculations are performed for each lifetime cost described above and then summed into the total Levelized Cost of Energy for each solution.

Formula for each lifetime cost (for example electricity, district heat or investment cost of AHU):

$$LCOE_{Lifetime\ cost_x} = \frac{Lifetime\ cost_x}{Annual\ heating\ and\ cooling\ demand * Cooling\ equipment\ lifetime}$$

3.7 Sensitivity analysis

The LCOE calculations are based on parameters which are again based on present value, historic data or forecast. This makes the calculations valid only for those exact conditions, and gives no indication on how the LCOEs would be affected if one or several of the parameters were to change.

The parameters can be affected by several factors, both global and local. On the global scale, climate change may change the heating and cooling demand, while geopolitics can impact the interest rate, exchange rates, electricity prices and lead to increased subsidies on solutions with positive climate impact. As the local climate conditions may fluctuate even on short distances, the heating and cooling demand might change based on where in the city the building is placed. Economic factors may also be subject to change for each building project, based on the deals are made with the distributor of equipment and the interest rate obtained from the bank.

A sensitivity analysis is performed to highlight the weakness of the model and how a change one of the mentioned parameters would affect the LCOEs. The sensitivity analysis includes both parameters that to some extent cannot be controlled, and some that can be. The “uncontrollable” parameters are the heating and cooling demand, interest rate and electricity prices. The other parameters are the power and energy price on district heat used for cooling, and an investment subsidy on heat-driven cooling equipment.

4 Results & discussion

4.1 Performance of the projected reference buildings customized for ventilation cooling

The annual cooling and heating demand for the different solution, for all three cases are given in figure 4.1. The cooling demand do not include demand at temperatures below 18 degrees Celsius (rounded), as this demand is considered to be covered by free cooling. As can be seen from the figure, the increased heat recovery of the desiccant machine drastically reduces the heating demand, while the cooling demand stays more or less constant. Another important point is that the cooling demand is greatly exceeded by the heating demand, especially in the LEB and 2x LEB cases.

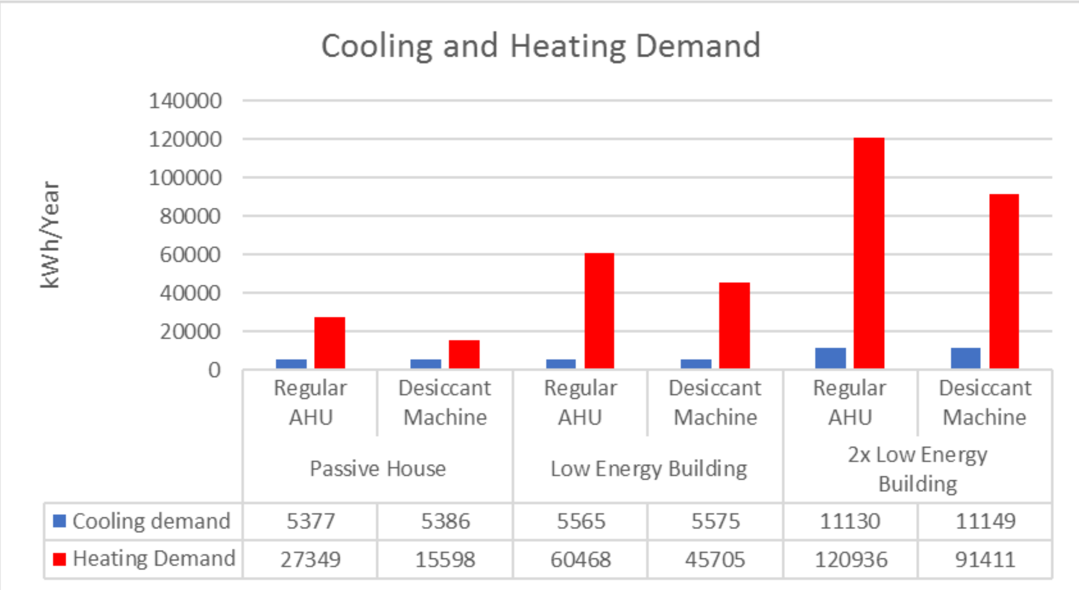

Figure 4.1: Cooling and heating demand in the different building cases with regular AHU or desiccant machine.

4.2 Results from the performance and cost calculation model

The electricity and district heat consumption, and the LCOE of the different heating and cooling solutions in each building case is collected and presented in section 4.2.1 and 4.2.2.

4.2.1 Electricity and District Heat consumption

Figure 4.2 presents the electricity and district heat consumption of the heating and cooling solutions for each building case. To visualize where the consumption originates from, the electricity consumption is separated into electricity used for air handling and electricity used for heating and cooling, while the district heat consumption is separated into district heat used for heating and district heat used for cooling.

The consumption patterns is similar for all three building cases. Since all solutions except the desiccant solution use the same AHU, the amount electricity used for air handling is naturally equal for all solutions except for the desiccant which has a slightly higher consumption. The electricity consumption used for heating and cooling on the other hand, is significantly reduced from the heat pump solutions to the electric chiller, and the heat driven solutions with close to zero consumption.. For solutions using district heat as heating source, the COP is considered to be 1, meaning that the consumption is equal for all cases except for the desiccant solution, which has a lower heating demand due to its enhanced heat recovery. When it comes to the district heat used for cooling, the absorption cooler has a much larger consumption of district heat than the desiccant machine due to its low COP.

One can see that by replacing electricity driven heating or/and cooling solutions with heat driven solutions, some of the electricity consumption can be replaced by low grade thermal energy. Comparing the numbers across cases, one can see that the difference increase from the PH case with the lowest heating and cooling demand to the 2x LEB with the largest demand. However, the amount of energy used is set to increase due to the low COPs compared to the electrically driven solutions.

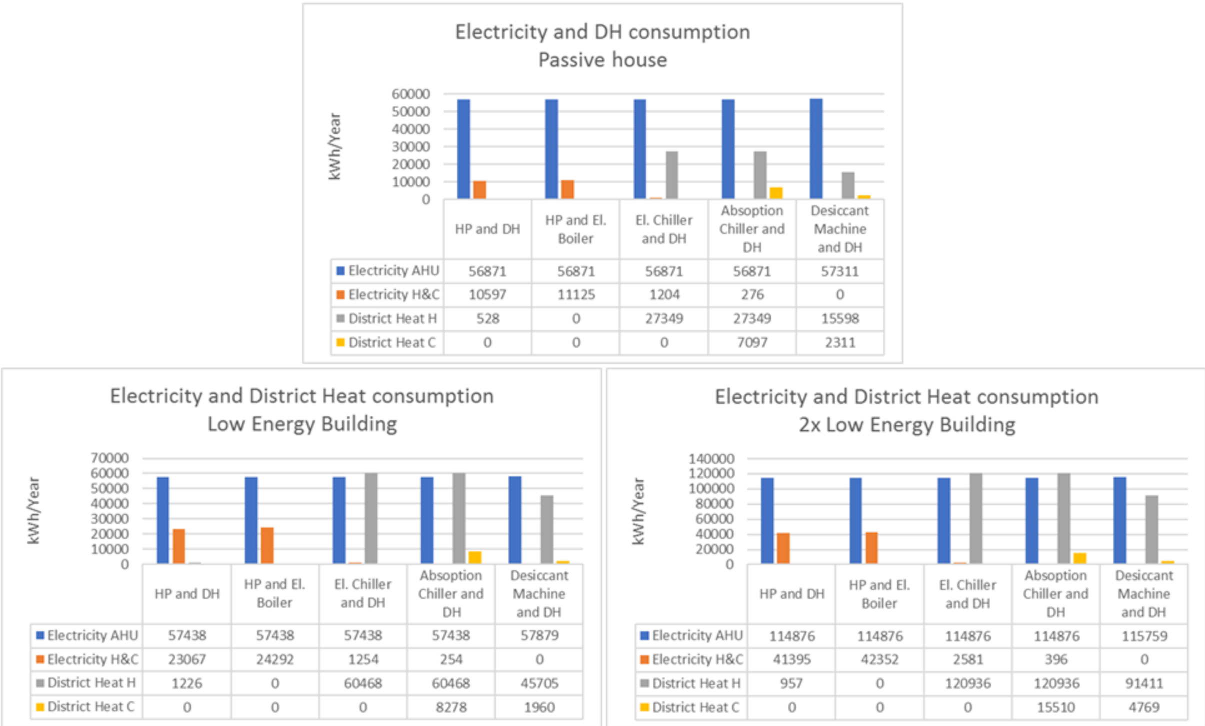

Figure 4.2: Electricity and District heat consumption of the different heating and cooling solutions in each case.

4.2.2 Levelized Cost of Energy

The heating and cooling solutions are benchmarked economically for each building case using the Levelized Cost of Energy method described in section 3.6.5. The LCOE result shows the NOK per kWh of heating and cooling demand satisfied through the lifetime of the cooling equipment. All LCOEs are calculated based on the heating and cooling demand of the solutions with regular AHUs, to avoid penalizing the desiccant solution for having a reduced heating demand due to its enhanced heat recovery.

Passive House

In the Passive House case, the desiccant solution is the solution with the lowest LCOE, while the absorption cooling and electric chiller solutions are the ones with the highest LCOE. Even though the desiccant solution has a high investment cost, the enhanced heat recovery makes it the most favorable option economically. The reason for the electric chiller and the absorption cooling solution to be more expensive than the heat pump solutions, is the low COP of district heat for heating. Additionally, the absorption cooler pays full price for the district heat, which is not viable with a COP much lower than the other solutions.

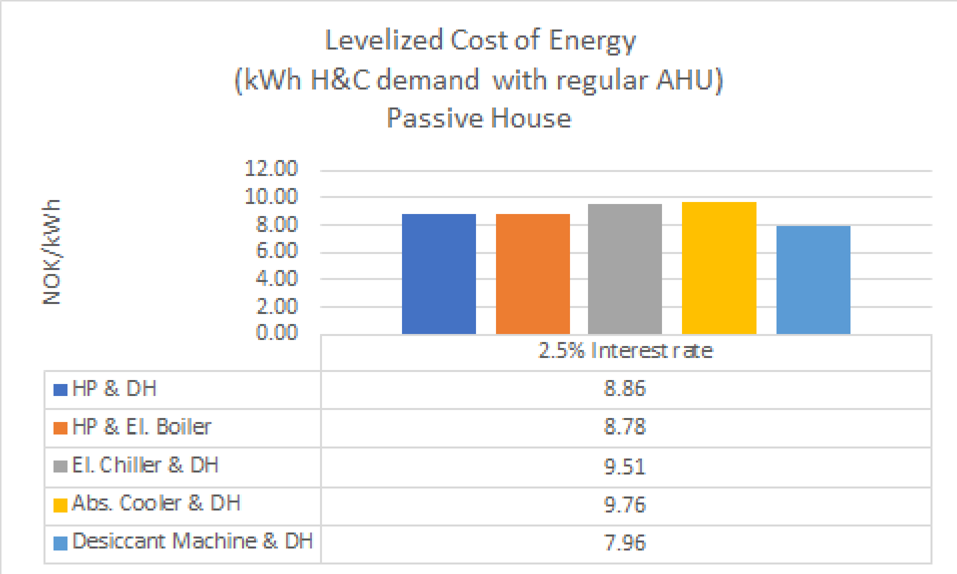

Figure 4.3: Levelized cost of Energy for the different heating and cooling solutions in the Passive House case.

Low Energy Building

The LCOEs for the Low Energy Building have similar tendensies as the ones in the PH case, but here the heat pump solutions have the lowest LCOEs, while the electric chiller and absorption cooler solutions are still the ones with the highest. The main cause of the difference in LCOEs from the PH case, is the increased heating demand. This favours the heat pumps, which has the highest COP on the temperatures used in the simulations (fairly warm winters).

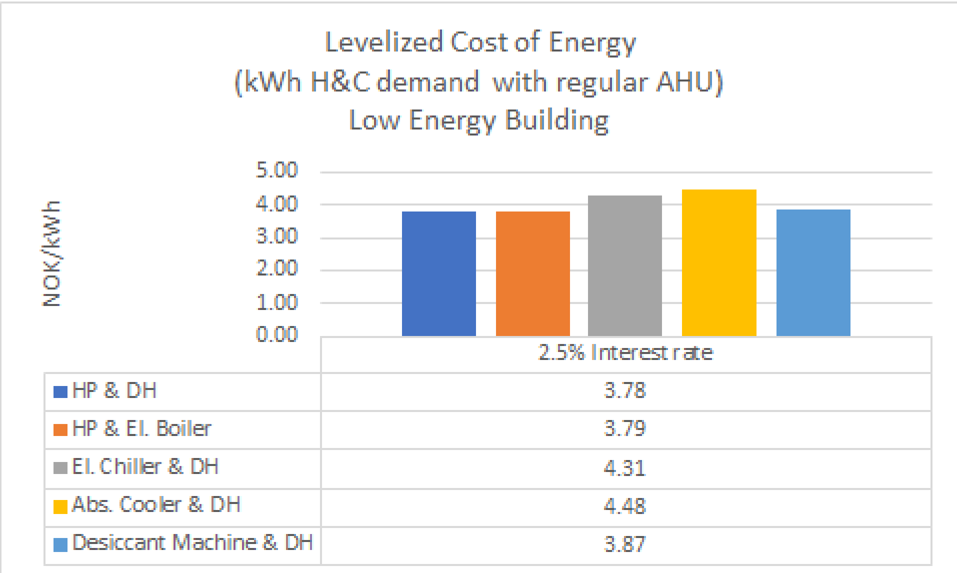

Figure 4.4: Levelized cost of Energy for the different heating and cooling solutions in the Low Energy Building case.

2x Low Energy Building

As figure 4.5 shows, there is one large difference in the LCOEs for the 2x LEB compared to the PH and LEB. The desiccant solution is the one with the highest LCOE. The difference is largely due to the increased investment costs when installing two desiccant machines, instead of one larger machine, which is the case for the other solutions.

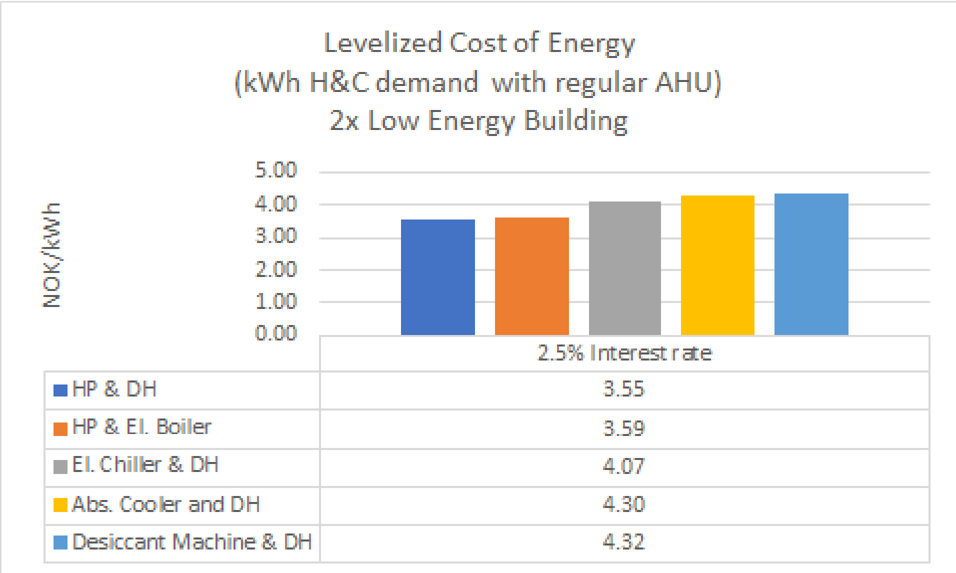

Figure 4.5: Levelized cost of Energy for the different heating and cooling solutions in the 2x Low Energy Building case.

4.2.3 Sensitivity Analysis

Figure 4.6 and figure 4.7 shows the results of the sensitivity analysis described in section 3.7. The parameters varied in analysis are the interest rate, electricity price, heating demand, cooling demand, a removal/reduction of the power and energy price on district heat used for cooling and a subsidy of the investment cost of heat driven cooling technologies.

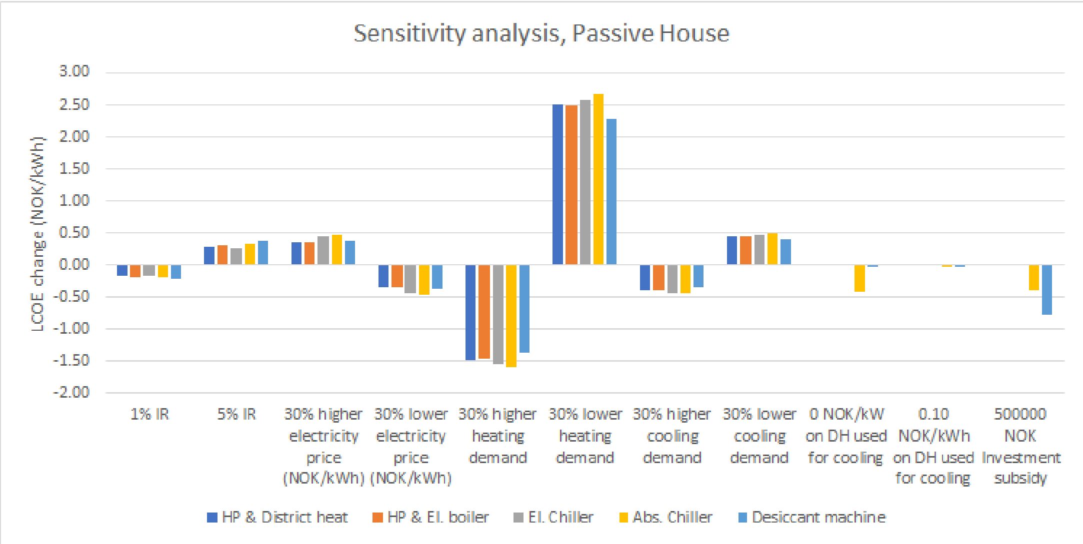

Figure 4.6: Sensitivity analysis for the Passive House case.

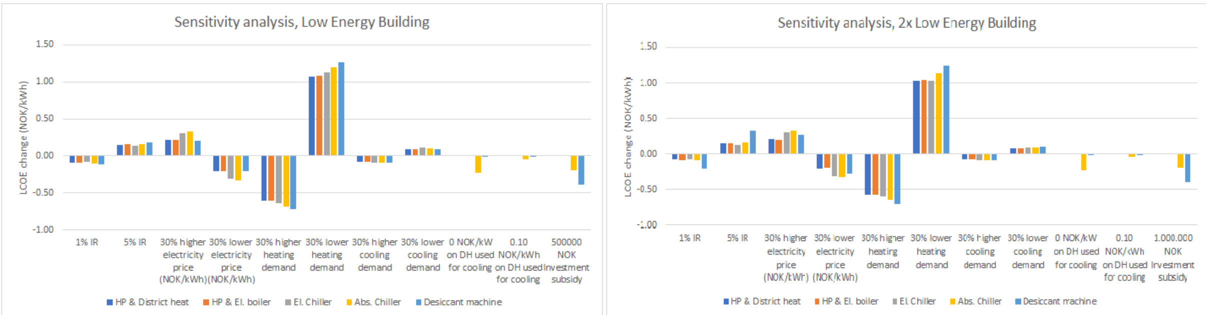

Figure 4.7: Sensitivity analysis for the LEB and 2x LEB case.

As expected, a reduced interest rate benefits the solutions with the highest investment cost over the lifetime of the cooling equipment, while increasing the interest rate penalize the same solutions. The solutions which is impacted the most is the desiccant cooling solution, while it has the lowest impact on the electric chiller solution. For the 2x LEB case, the magnitude of impact on the desiccant cooling solution increases significantly compared to the other solutions, because this solution includes two similar machines instead one larger.

Since the DH price is connected to the electricity price, increasing the electricity price also increases the DH price. This favors the solutions with low energy consumption, which is the heat pumps because of high COP and the desiccant solution because of its high heat recovery.

For the heating demand, the figures show that while the desiccant cooling solution has the least benefit/disadvantage in the passive house case, it has the largest benefit/disadvantage for the LEB and 2x LEB cases. What the figure do not show is that the percentage increase of the LCOE for the desiccant solution is the same for all cases, but since the heating demand of the PH with desiccant cooling is so low, it does not benefit in the same magnitude in terms of NOK/kWh of heating and cooling demand. One can also see that the heat pump solutions have a slight difference in the change in LCOE, which implies that the peak load solutions are used. Since the electricity price is slightly higher than the DH price, the solution with electric boiler will have a slightly higher energy cost even though the COP is 1 for both the boiler and DH. This is also the reason behind the decrease in heating demand has a higher effect on the heat pump and electric boiler than on the electric chiller solution. However, the peak power cost may affect the heat pump solutions in the other direction. While the electric boiler power comes on top of the electricity used by the heat pump, the district heat power is charged separately. This means that while a peak load demand at -3 C might have the highest energy load, a peak load demand at -4 C might have a lower energy load, but since the capacity of the heat pump is reduced at lower temperatures, the fraction of district heat is higher than in the first case. This would lead to a higher power cost in the HP & DH case, and explains why the solution is more affected by the heating demand increase in the passive house case, even though the energy consumption is equal and the energy price is lower.

The electric chiller is more affected by the increased and decreased cooling demand than the heat pumps. This is because, at least for the equipment used in these cases, the COPs of the electric chillers are lower than the ones of the heat pumps. The absorption chiller is not affected as much as might be expected considering its low COP. The reason is that it always runs full capacity and shuts off when the demand is covered, which means that an increase in the heating demand would not change the peak power cost, as it does for the other solutions. The desiccant solution has a higher COP and is not heavily affected either. Since the cooling demand for the three building cases are relatively low compared to the heating demand, an equal demand increase in percentage would not be close of having the same impact on the LCOE as the heating demand.

When it comes to the controllable parameters, both the removal of the power price and the reduction of the energy price on district heat used for cooling affects the absorption cooling case the most. This is a result of a lower COP and that the Absorption chiller always runs on full capacity. These two factors lead to a higher energy consumption and peak power compared to the desiccant machine who has a higher COP and regenerates the desiccant wheel based on demand. Additionally, the power and consumption of the desiccant machine are based on an average COP described in section 3.5.6, which may have a lowering effect on its peak power. The investment subsidy on the other hand, has about twice the effect on the desiccant solution compared to the absorption chiller solution. This is because the desiccant machine's lifetime is half compared to the lifetime of the absorption chiller. As a result, the subsidy would be paid twice for the desiccant machine and once for the absorption chiller, over the lifetime of the absorption chiller.

4.2.4 Scenarios with investment subsidy and reduced price on district heating used for cooling

To illustrate how investment subsidies and reducing the power and energy price on district heat would affect the competitiveness of the heat driven solutions, seven scenarios were created in addition to the one in section 4.2.2. These scenarios consist of introducing the investment subsidy on heat driven cooling technologies, price reduction and both at 2.5% and 5% interest rate. As pointed out in the sensitivity analysis, the price reduction has minor effect on the desiccant cooling solutions, but it is included in the figure for illustration.

The investment subsidy was set to 500000 NOK for the PH and LEB cases, and doubled for the 2x LEB case since it is in fact two low energy buildings. For the price reduction, the price on power was removed for DH used for cooling, while the energy price was set to 0.10 NOK/kWh, which is a significant reduction from the regular monthly prices estimated in section 3.3.

Figure 4.8, 4.9 and 4.10 show the LCOEs of the different solutions for each scenario. It can be seen that the doubled interest rate does not have a large enough impact on the competitiveness of the different solutions to make a difference in the choice of solution. Without any subsidy or price reduction on DH, the desiccant solution can compete economically in the PH case, and is almost level with the HP solutions in the LEB case as well. The absorption cooling solution on the other hand, is not able to compete without subsidy or price reduction.

The absorption chiller can compete against the heat pump solutions in the PH case, if both price reduction and investment subsidy is introduced. In the other scenarios, it is able to compete with the electric chiller solutions either by introducing the investment subsidy or the price reduction on DH for cooling. However, it is not able to compete with the desiccant cooling solution, except in the 2x LEB where two desiccant machines are bought, leading to the high investment cost.

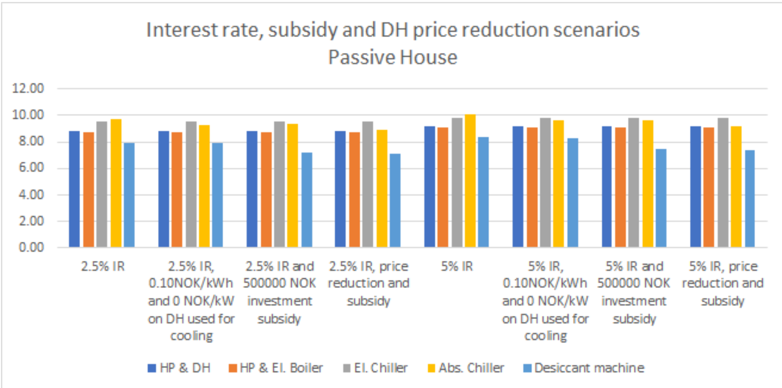

Figure 4.8: LCOE scenarios for the passive house case.

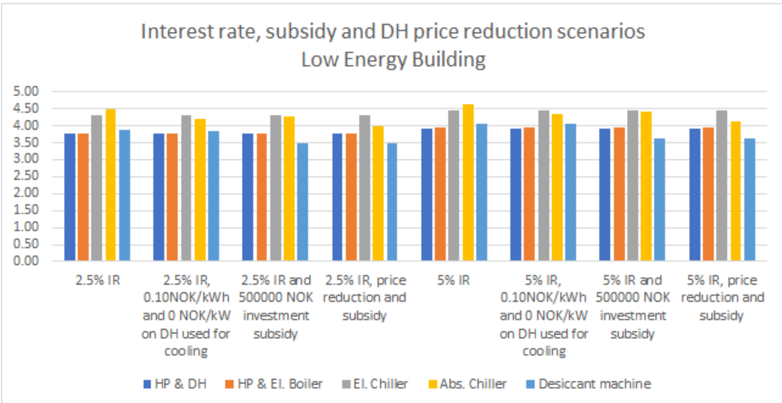

Figure 4.9: LCOE scenarios for the Low Energy Building case.

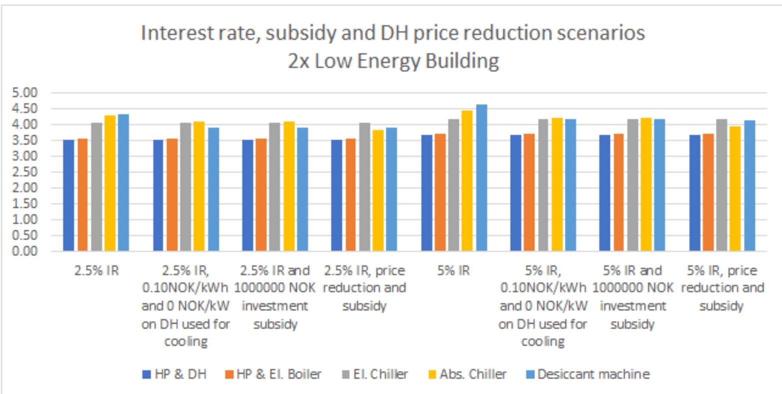

Figure 4.10: LCOE scenarios for the 2x Low Energy Building

4.3 Sustainability discussion

The levelized cost calculations and the additional scenarios show that the heat driven cooling solutions can be economically sustainable even without investment subsidy. The desiccant solution especially, because of its improved heat recovery. The absorption cooling solution on the other hand require a reduction of the energy and power price on the district heating used for cooling to be able to compete with the electric chiller solution, and is not able to compete with the heat pumps in the current climate without investment subsidy.

In terms of the environmental sustainability, the heat driven cooling solutions are free of the environmental unfriendly refrigerants used in the heat pumps and electrical chillers (Aermec) (Thermo Control, 2018) (Yazaki Nordic) (Ghassem Heidarinejad, 2010). They also switch electricity consumption over to district heat, which releases capacity in the electricity grid. Since the thermal energy used for cooling is otherwise wasted, the increase in the total energy demand can be considered to have no additional polluting effect, only removing the potential pollution created by the avoided electricity consumption.

The situation changes if both boilers in the waste to energy plant go down at the same time or the plant is shut down. If the district heating is produced by an electric boiler with a COP of 1 or below, all of the district heat demand would switch to electricity. The increase in total energy demand would then have a negative impact, as it would increase the electricity consumption and the load on the electricity grid. However, to opt for such situations, Tafjord has installed a 7000 m^3 accumulator tank, that is estimated to cover two days of district heating demand during winter time. As the consumption of district heat is lower during the summer, this time estimate can be considered low if the boilers go down during summer time (Irene Vik, 2017).

5 Conclusion

The work covers the implementation and benchmarking of heat driven versus electrically driven cooling and heating solutions in buildings projected according to the energy requirements of the building regulations and the Passive House and Low Energy Building criteria.

Projecting buildings in Aalesund according to the latest TEK building regulations leaves little room for flexibility. If the LEB was to be projected solely according to the LEB criteria, the total net energy demand of the building would not satisfy the TEK17 building regulations. The energy demand requirement of TEK17 is the constraining factor because it must be calculated using Oslo climate and normative values for technical equipment and lighting, even if the building is built in Aalesund and with internal loads of technical equipment and lighting according to PH and LEB criteria. With the internal load requirements of the PH and LEB criteria being stricter than TEK17 and Aalesund having a milder climate, the TEK17 energy requirements are artificially high. This means that the building requirements are stricter for Aalesund than for Oslo, limiting the heating and cooling requirement of projected buildings.

The heating and cooling demand and the energy consumed by the projected buildings shows that for buildings projected after current regulations and standards, the heating demand is dominant in terms of energy required. This benefits the solutions with high heating COP and limits the effect of reducing the price on district heat consumed for cooling.

The LCOE analysis shows that while the desiccant cooling machine is barely affected by a price reduction on district heat power and consumption, the price reduction plays a significant role in making the absorption cooler competitive. This is because the absorption cooler has a much higher consumption of district heat and a high thermal power input while running, making the power cost substantial regardless if it is operated for one minute or hundred hours during the month. Investment subsidies would naturally affect the LCOE of both solutions, especially if the interest rate increases. If the sum of subsidy would be fixed, the desiccant solution would have the largest benefit in term of LCOE because of the lower lifetime.

The desiccant solution could be competitive against heat pump solutions even without subsidies or price reduction on district heat for cooling. This is mainly because of its enhanced heat recovery reducing the heating demand. The absorption cooler solution on the other hand requires both subsidy and price reduction to come close to the heat pump solutions in terms of LCOE. However, either one could make it competitive against the electric chiller solutions, which it would be natural to compare it with, since both solutions use district heat to cover the heating demand. With increasing heating demand the solutions using district heat as their heat source, become less competitive due to the heat pump's high COP.

Other important factors that is not covered by the LCOE analysis is the reliability and environmental aspect. None of the heat driven cooling solutions use any environmentally unfriendly refrigerants, and the maintenance of the cooling machines are minor. The lifetimes of the machines are estimated to be 20 years for the desiccant and 40 years for the absorption cooler, compared to 15 years for the heat pumps and electric chiller. The heat driven cooling solutions can therefore be considered more reliable both in terms of regulations on refrigerants and on maintenance and lifetime. An additional important factor is that compared to the heat pump solutions, the heat driven cooling solutions with district heat to cover the heating demand can replace large quantities of electricity consumption with low-grade thermal energy.

Considering both the economic, environmental and reliability factors, the heat driven cooling solutions could be a viable option and should be considered when implementing heating and cooling equipment in buildings connected to the district heating network in Aalesund or other locations with similar climate.

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APPENDIX

SECTION 3.1.2

Table: Maximum total net energy demand for different building categories (Norwegian Building Authority, 2017).

Building category	*Total net energy requirement (kWh/m^2 heat gross internal area per year)*
Small houses and leisure homes with more than 150 m^2 of heated gross internal area	100 + 1.6/m^2 heated gross internal area
Block of flats	95
Kindergarten	135
Office Building	115
School	110
University	125
Hospital	225 (265)
Nursing home	195 (230)
Hotel	170
Sports building	145
Commercial building	180
Cultural building	130
Body shop	140 (160)

Table: Minimum requirements for building components according to TEK 17 (Norwegian Building Authority, 2017).

Attribute	*Minimum requirement according to TEK17*	*Unit*
U-value external walls	≤ 0.22	$W/(m^2 * K)$
U-value roof	≤ 0.18	$W/(m^2 * K)$
U-value floor on ground and facing open air	≤ 0.18	$W/(m^2 * K)$
U-value windows and entrance doors, including frames	≤ 1.20	$W/(m^2 * K)$
Air infiltration rate at 50 Pa	≤ 1.5	h^{-1} **(Air change per hour)**

Table: Cooling coefficient to determine maximum net specific cooling demand for PH/LEB (Standard Norge, 2012).

Building category	***Passive House***	***Low Energy Building***
	β	β
Kindergarten	0.75	0.75
Office Building	1.4	2.1
School	0.75	0.75
University	1.5	3.0
Hospital	2.9	3.6
Nursing home	1.6	2.3
Hotel	1.5	2.2
Sports building	0.9	1.6
Commercial building	3.3	4.8
Cultural building	1.2	1.9
Body shop	1.1	1.8

Table: Maximum calculated annual heating demand for PH/LEB buildings over 1000 m² in climates with annual mean temperature above 6.3 Celsius (Standard Norge, 2012).

Building category	***Passive House***	***Low Energy Building***
	$kWh/(m^2 \cdot year)$	$kWh/(m^2 \cdot year)$
Kindergarden	25	40
Office Building	20	35
School	20	30
University	20	35
Hospital	20	35
Nursing home	20	30
Hotel	25	40
Sports building	20	35
Commercial building	25	40
Cultural building	25	40
Body shop	25	40

Table: Internal gains during operating hours for net energy demand calculations (Standard Norge, 2012) (Norwegian Building Authority, 2017).

	Passive house	**Low Energy Building**	**TEK17 (NS 3031:2014)**	**Average load during operating hours**
Technical equipment	6	6	11	$\boldsymbol{W/m^2}$
Lighting	4	4	8*	$\boldsymbol{W/m^2}$
Persons	4	4	4	$\boldsymbol{W/m^2}$
*The lighting internal heat gain value for TEK17 calculation can be reduced if a sufficient demand control system is documented.				

SECTION 3.2.1

Figure: Building layout

Table: General building specifications that are similar for both the projected passive house and low energy building

Office hours	The office hours set the operation hours for the ventilation systems, heating system, internal loads and doors inside	**7 am – 7 pm**
Windows 1.4mx1.5m	All the office windows and two windows in the 1st floor cafeteria	**U-Value differs depending on whether the building is following the PH or LEB standard**
Windows 1.4mx2.1m	All windows at the entrance, all windows in the 2nd,3rd and 4th floor break room, and four out of six windows in the 1st floor cafeteria	**U-Value differs depending on whether the building is following the PH or LEB standard**
Entrance door 2mx2.2m	Located in the Entrance/reception on the 1st floor	**U-Value differs depending on whether the building is following the PH or LEB standard**
Windows 1.5mx1.55m / 1.5mx2.0	The window sizes in the 20.25m^3 offices vary between the versions. For the passive house the size is 1.5mx1.55m, while they are 1.5mx2.0m in the LEB	**U-Value differs depending on whether the building is following the PH or LEB standard**
Emergency exit doors 1mx2.1	All doors to the outside except the entrance door	**U-Value differs depending on whether the building is following the PH or LEB standard**
Doors inside 1mx2.1m	All the doors inside	**Open 25% of the time during office hours**
Openings without doors 3.3mx2.2m and 6.6mx2.2m	The openings between the corridors and the entrance/hallway are 3.3mx2.2m, while the opening between the Break rooms and the entrance/hallway are 6.6mx2.2m	**Always 100% open**
Indoor walls	Separation walls between similar offices	**U-Value: 0.25 (W/m^2K)**
Indoor walls	All other indoor walls	**U-Value: 0.15 (W/m^2K)**
Ceiling height	Same for all floors	**2.5 meter**

SECTION 3.2.2

Table: Projected attributes for building parts, components and air infiltration (Norwegian Building Authority, 2017) (Standard Norge, 2012).

Attribute	*Projected Passive house*	*Minimum requirement according to the Passive house criteria, TEK17 and NS 3031:2014*	*Unit*
U-value external walls	0.10	≤ 0.22	$W/(m^2 * K)$
U-value roof	0.08	≤ 0.18	$W/(m^2 * K)$
U-value floor	0.08	≤ 0.18	$W/(m^2 * K)$
U-value windows and entrance doors	0.80	≤ 0.80	$W/(m^2 * K)$
Air infiltration rate at 50 Pa	0.60	≤ 0.60	h^{-1} **(Air change per hour)**
Thermal bridge value	0.03	≤ 0.03	$W/(m^2 * K)$
Specific fan power (Ventilation)	1.49	≤ 1.5	$kW/(m^3/s)$
Ventilation heat recovery efficiency (yearly average)	0.82	≥ 0.80	**%**

Table: Performance of the projected Passive House (Norwegian Building Authority, 2017) (Standard Norge, 2012).

Attribute	*Projected Passive House*	*Minimum requirement according to the Passive house criteria, TEK17 and NS 3031:2014*	*Unit*
Maximum cooling load	89.4*		kW
Cooling demand /yr	3.4** (13527)	≤ 3.4	$kWh/m^2/yr$ (kWh/yr)
Maximum heating load (Space heating)	132.5*		kW
Heating demand /yr (space heating)	6.6** (26551)	≤ 20	$kWh/m^2/yr$ (kWh/yr)
Heating demand /yr (water heating)	5.0** (20046)		$kWh/m^2/yr$ (kWh/yr)
Total net specific energy demand (Passive house / Low energy building evaluation)	63.4**		kWh/m^2 $(heated\ gross\ area)$
Total net specific energy demand (TEK17)	97.4***	≤ 115	kWh/m^2 $(heated\ gross\ area)$
Heat loss rate	0.22**	≤ 0.40	$W/(m^2 * K)$
*From summer/winter simulation in SIMIEN. **From passive house evaluation in SIMIEN. ***From evaluation against TEK17 in SIMIEN.			

Table: Projected attributes for building parts, components and air infiltration (Norwegian Building Authority, 2017) (Standard Norge, 2012).

Attribute	*Projected Low Energy Building*	*Minimum requirement according to the Low Energy Building criteria, TEK17 and NS 3031:2014*	*Unit*
U-value external walls	0.15	≤ 0.22	$W/(m^2 * K)$
U-value roof	0.10	≤ 0.18	$W/(m^2 * K)$
U-value floor	0.10	≤ 0.18	$W/(m^2 * K)$
U-value windows and entrance doors	1.20	≤ 1.20	$W/(m^2 * K)$
Air infiltration rate at 50 Pa	1.5	≤ 1.5	h^{-1}
Thermal bridge value	0.05	≤ 0.05	$W/(m^2 * K)$
Specific fan power (Ventilation)	1.49	≤ 2.0	$kW/(m^3/s)$
Ventilation heat recovery efficiency (yearly average)	0.82	≥ 0.70	**%**

Table: Performance of the projected Low Energy Building (Norwegian Building Authority, 2017) (Standard Norge, 2012).

Attribute	*Low energy building*	*Minimum requirement according to the Low Energy Building criteria, TEK17 and NS 3031:2014*	*Unit*
Maximum cooling load	110.0*		kW
Cooling demand /yr	4.9** (19594)	≤ 5.0	$kWh/m^2/yr$ (kWh/yr)
Maximum heating load (Space heating)	168.1*		kW
Heating demand /yr (space heating)	14.8** (59023)	≤ 35	$kWh/m^2/yr$ (kWh/yr)
Heating demand /yr (water heating)	5.0** (20046)		$kWh/m^2/yr$ (kWh/yr)
Total net specific energy demand (Passive house / Low energy building evaluation)	73.6**		kWh/m^2 $(heated\ gross\ area)$
Total net specific energy demand (TEK17)	112.3***	≤ 115	kWh/m^2 $(heated\ gross\ area)$
Heat loss rate	0.37**	≤ 0.50	$W/(m^2 * K)$
*From summer/winter simulation in SIMIEN. **From passive house evaluation in SIMIEN. ***From evaluation against TEK17 in SIMIEN.			

Table: Dimensioning maximum loads for heating and cooling. Based on summer and winter simulations.

Attribute	*Projected Passive House*	*Projected Low energy building*	*2x Projected Low energy building*	*Unit*
Maximum cooling load	89.4	113.9	227.8	***kW***
Maximum heating load (Space heating)	106.5	174.5	349	***kW***

SECTION 3.5.1

Table: Cost and lifetime estimate for the air handling unit (Andersen, 2018) (K. Lindsetvik, 2018).

	One AHU	**Four AHUs**	**Eight AHUs**	
Investment cost	209 188	836 750	1673500	NOK
Annual service cost		66 092	132 184	NOK/year
Lifetime	20	20	20	Years

SECTION 3.5.2

Table: Investment cost and lifetime estimate for the accumulator tanks (A. R. Engen, 2018).

	VKG 1500	**VKG 3000**	
Investment cost	31 250	52 500	NOK
Lifetime	15	15	Years

SECTION 3.5.3

Table: Cost and lifetime estimate for the heat pump and boiler in the passive house building case (A. R. Engen, 2018).

	Heat pump	**El. Boiler**	
Investment cost	281 250	90 000	NOK
Annual service cost	6000	0	NOK/year
Lifetime	15	15	Years

Table: Cost and lifetime estimate for the heat pump and boiler in the low energy building case (A. R. Engen, 2018).

	Heat pump	El. Boiler	
Investment cost	330 000	140 000	NOK
Annual service cost	6000	0	NOK/year
Lifetime	15	15	Years

Table: Cost and lifetime estimate for the heat pump and boiler in the 2x low energy building case (E. Løberg, 2018).

	Heat pump	El. Boiler	
Investment cost	593 750	156 250	NOK
Annual service cost	20 000	0	NOK/year
Lifetime	15	15	Years

SECTION 3.5.4

Table: Cost and lifetime estimates for the absorption and dry cooler for each case (A. Nesthorne, 2018).

	Abs. cooler LEB and Passive house	Abs. Cooler 2x LEB	Dry cooler LEB and Passive house	Dry Cooler 2x LEB	
Investment cost	636 048	1 381 200	198 000	336 000	NOK
Annual service cost	750	750	0	0	NOK/year
Lifetime	40	40	15	15	Years

SECTION 3.5.5

Table: Cost and lifetime estimate of the electric chiller in each case (T. Lie, 2018).

	Electric chiller Passive house	Electric chiller LEB	Electric chiller 2xLEB	
Investment cost	200 000	237 500	400 000	NOK
Annual service cost	10 000	10 000	10 000	NOK/year
Lifetime	15	15	15	Years

SECTION 3.5.6

Table: Relative humidity at Vigra weather station (local airport). The table shows the average percentage amount of time with X% relative humidity at each temperature step (2004-2017) (eKlima, 2018).

Relative humidity (rounded)	18 °C	19 °C	20 °C	21 °C	22 °C	23 °C
5%	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %
15%	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %
25%	0,0 %	0,1 %	2,4 %	1,3 %	1,6 %	1,1 %
35%	2,4 %	6,5 %	8,9 %	6,7 %	6,3 %	9,5 %
45%	7,1 %	9,7 %	14,2 %	18,2 %	36,7 %	33,7 %
55%	19,8 %	23,3 %	28,2 %	33,8 %	32,0 %	31,6 %
65%	25,4 %	28,4 %	26,6 %	27,6 %	18,8 %	23,2 %
75%	28,9 %	26,2 %	18,9 %	12,0 %	4,7 %	1,1 %
85%	15,0 %	5,4 %	0,8 %	0,4 %	0,0 %	0,0 %
95%	1,4 %	0,3 %	0,0 %	0,0 %	0,0 %	0,0 %
100%	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %	0,0 %

Table: Cost and lifetime estimate of the desiccant cooling machine in each case (A. Grandstrand, 2018).

	Desiccant cooling machine PH and LEB	Desiccant cooling machine 2x LEB	
Investment cost	1 777 656	3 555 312	NOK
Annual service cost	72 701	145 402	NOK/year
Lifetime	20	20	Years

SECTION 3.6.3

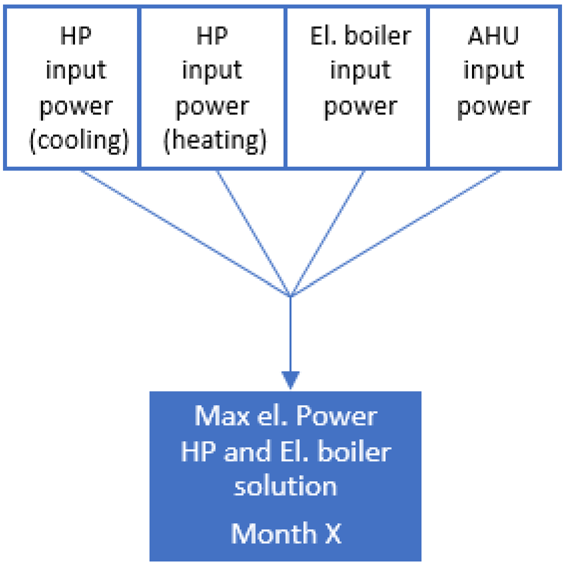

Figure: How the hourly value parameters are distributed and used for peak power calculations in the heat pump and el. boiler solutions.

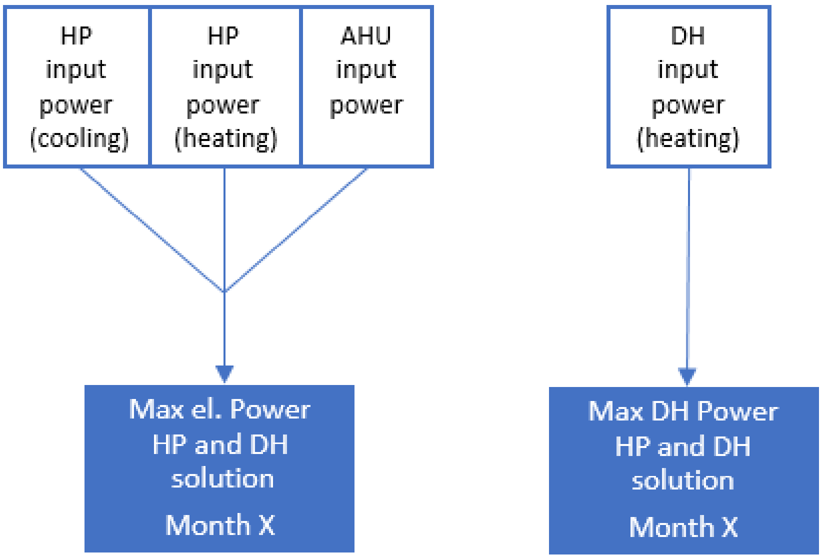

Figure: How the hourly value parameters are distributed and used for peak power calculations in the heat pump and district heat solutions.

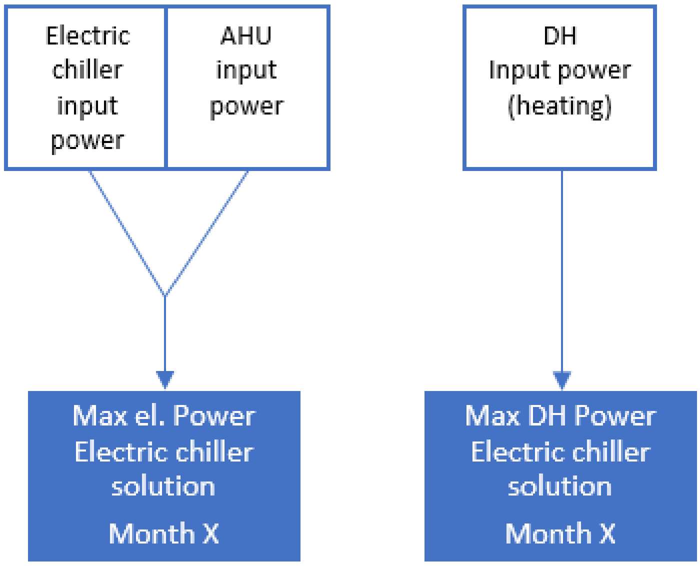

Figure: How the hourly value parameters are distributed and used for peak power calculations in the electric chiller solutions.

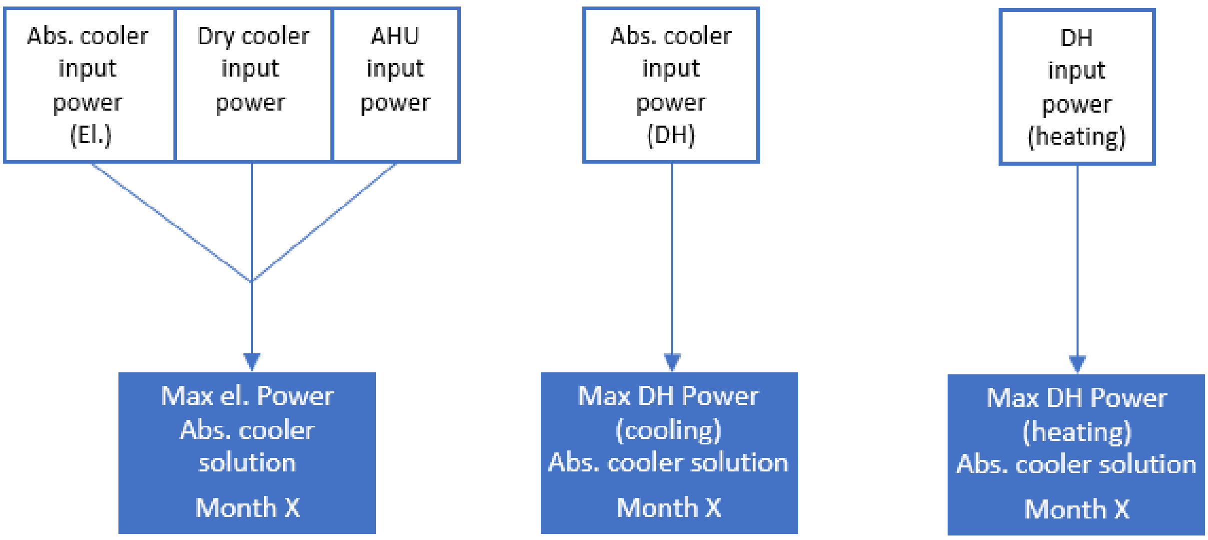

Figure: How the hourly value parameters are distributed and used for peak power calculations in the absorption cooler solutions.

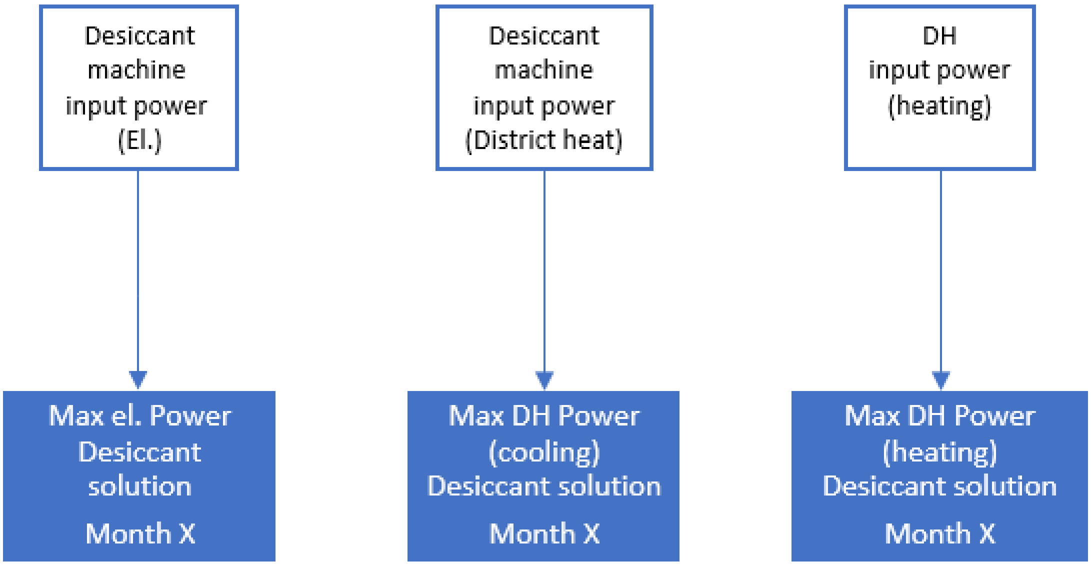

Figure: How the hourly value parameters are distributed and used for peak power calculations in the desiccant machine solutions.